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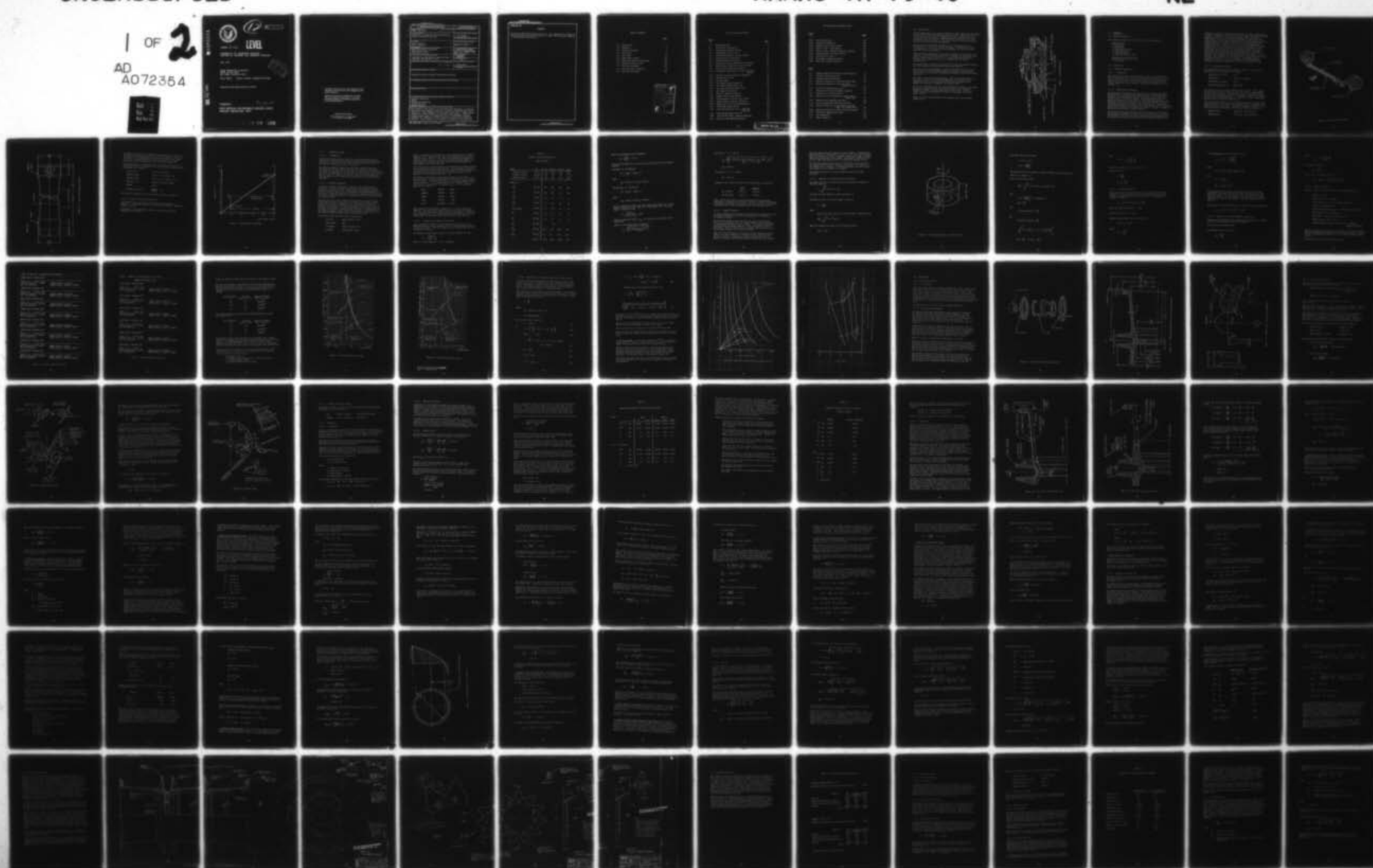

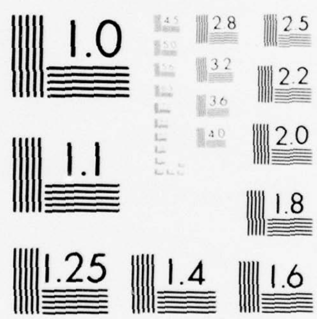

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LEVEL II

AMMRC TR 79-40

FEASIBILITY OF COMPOSITE MATERIAL APPLIED TO TRACKED AND WHEELED VEHICLES

June 1979

Riggs Engineering Corporation
8245A Ronson Road
San Diego, California 92111

Final Report Contract Number DAAG46-78-C-0066

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Prepared for

ARMY MATERIALS AND MECHANICS RESEARCH CENTER
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The purpose of this contract is to determine the feasibility of applying fiber reinforced composite materials to tracked and wheeled Army vehicle components. Seven components are included in the study. Several design concepts are shown together with load and stress analysis. Material properties are given. The projected cost of a development program is presented for components for which composite materials were determined feasible. Also a quantity cost estimate is projected to determine the

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ABSTRACT

savings to fabricate different quantities. The components are summarized by a prioritized listing indicating cost, weight and performance compared to the baseline (metal) parts.

TABLE OF CONTENTS

List of Figures and Tables

List of Figures and Tables (Cont.)

1.0 Introduction

This document reports on a program conducted by Riggs Engineering Corporation for the Army Materials and Mechanics Research Center. The program was for a study of seven Army vehicle components, presently fabricated in steel, to determine the feasibility of fabricating these components from fiber reinforced composite materials. The study included a design and analysis as well as load and envelope requirements.

The program was divided into two phases. The first phase was the establishment of a baseline for each component. This baseline becomes the requirements for loads, envelope, performance, as well as the base reference for cost and weight.

The second phase consisted of the feasibility study for the design of the component in fiber reinforced composites. Candidate fiber materials were glass, graphite, Kevlar, and boron. The boron fiber however was not utilized in any component due to cost, manufacturing, or design considerations.

The design study included not only material, stress, and envelope studies, but also manufacturing methods and tooling considerations.

Under the manufacturing analysis of each part determined feasible, the projected cost of a development program is estimated. Also a quantity cost estimate is projected to determine the savings to fabricate different quantities.

The report is divided into two sections. The first section reports on the study of each of the individual components. The second section summarizes the components by a prioritized listing indicating cost, weights and performance compared to the baseline parts.

Of the seven components only two were considered not feasible, the torsion bar and the connecting link. Two other components were considered marginally feasible, the road wheel and the idler wheel. Because of impact loads and interchangeability these two components show little weight savings.

Figure 1-1 shows the M-60 Combat Tank components which were included in the study.

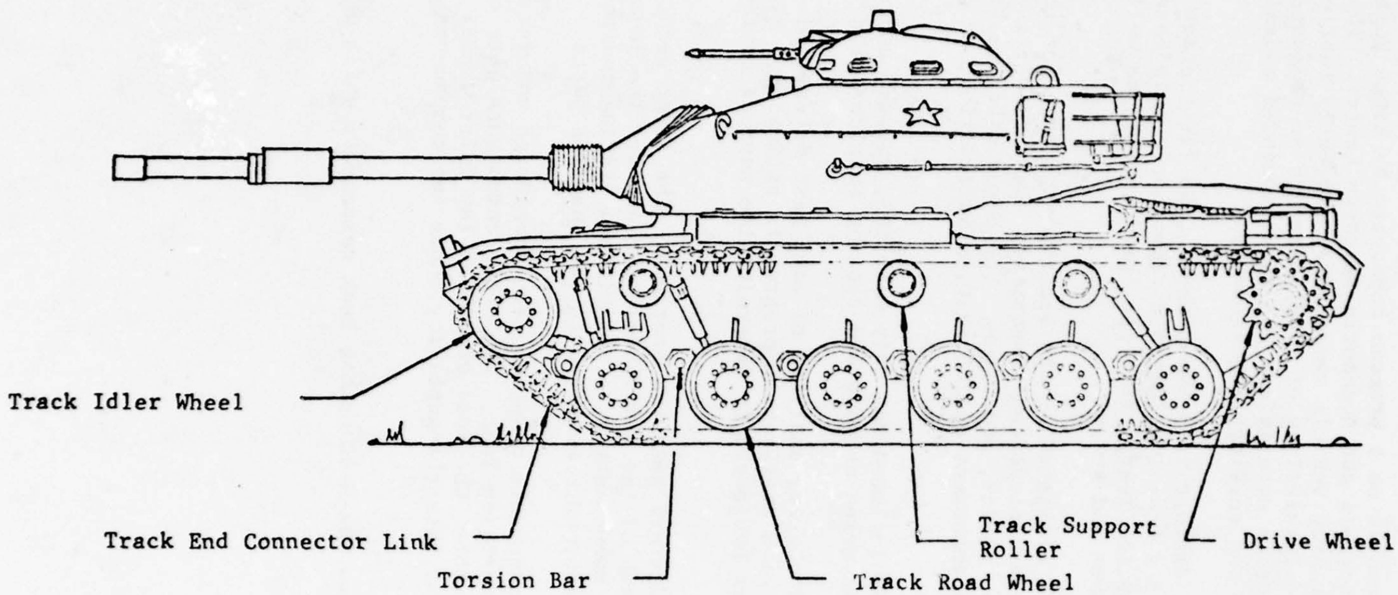


Figure 1-1 M-60 Combat Tank

2.0 COMPONENTS

2.1 SELECTED COMPONENTS

The following components were selected for the study program.

M-60 Army Tank

1. Torsion Bar
2. Drive Wheel
3. Track Support Roller
4. Track Idler Wheel
5. Track Road Wheel
6. Track End Connector Link

Prototype Five Ton Army Truck

7. Truck Wheel

2.2 TORSION BAR

2.2.1 Baseline Requirements

2.2.1.1 Description

The torsion bars are part of the suspension system on the M-60 Combat Tank. They provide the necessary spring force to each road wheel. The torsion bar is mounted to the swing arm which supports the road wheel and runs across the width of the vehicle where it is mounted to the opposite road wheel support housing. The general arrangement is shown in Figure 2-1.

2.2.1.2 Dimensional Requirements

The dimensional requirements were determined from Drawings No. 8668989, Spring Torsion Bar Suspension and 10905415 sheet 1 to 3, Suspension Assembly. The first drawing provides the principal dimensions of the torsion bars No. 7359890 and 7359891. The second drawing shows the method of joining the torsion bar end to the road wheel swing arm. The difference between the torsion bars No. 7359890 and 7359891 is in the direction of the pre-set, otherwise they are identical. Consequently, the composite design of only one of the torsion springs will be considered. Twelve (12) torsion bars are required per vehicle. Figure 2-2 shows the dimensions of the present torsion spring bar.

The torsion bars are mounted to the vehicle through bolted on housings which contain the anchor (or fixed) end of one torsion bar, as well as the bearing (or rotating) end of the torsion bar for the opposite wheel. The anchor end is shown in Section H-H and the bearing in detail A of

Drawing No. 10905415. Detail A on sheet 3 and the vehicle cross section on sheet 1 indicate that the housing is quite long. The effective housing length was determined by scaling the drawing and was estimated as approximately 22 in. from the face of the large end of the bar. The inside diameters of the wheel arm bearing and of the housing were scaled as approximately three inches. This leaves a clearance between the shaft and the wall of only (3-2.35) x .5 = .33 in. Consequently the external diameter of either ends of the torsion bar cannot be increased significantly. At the center of the vehicle, an enlargement of the torsion bar may be possible, but would require a check of potential interference with the vehicle systems and floor body. For design purposes it was assumed, that the present diameter of 2.35 in. is the desirable outer dimension and may be increased, if necessary, to 2.70 in. maximum. This limitation is determined by the inside spline diameter of the large end, which is 2.743 in. minimum. The maximum diameter envelope must also include any fastener which may be required for joining the composite torsion bar to metallic serrated end fitting.

2.2.1.3 Design Load Condition

The design load is specified in Appendix D, Table I, Item (1) of the Specifications and Requirements as follows:

Maximum Angle of Twist	50.5°
Spring Rate	7,330 in. lbs/degree
Road Wheel Arm Length	16 in.
Fatigue Requirements per	MIL-S-45387

The spring torsion bar drawing (8668989) also specifies, that a preset test is required, twisting the bar three times to 83°. The permanent set should not exceed 30°. The maximum allowable windup angle should be 50.5° and the maximum allowable set angle after fatigue tests between 7° and 49° twist should not exceed 5°. However the preset requirement may not apply to a fiberous composite bar.

With the road arm length of 16 in. the road wheel load per degree twist of the torsion bar is 7330/16 = 458 lbs/degree and at the maximum twist angle 458 x 50.5 = 23,135 lbs. According to Appendix D, Table I, of the Specifications and Requirements, Item (5) the road wheel loads are:

Maximum Static	11,250 lbs. (left #3 wheel)
Minimum Static	4,075 lbs. (right #6 wheel)

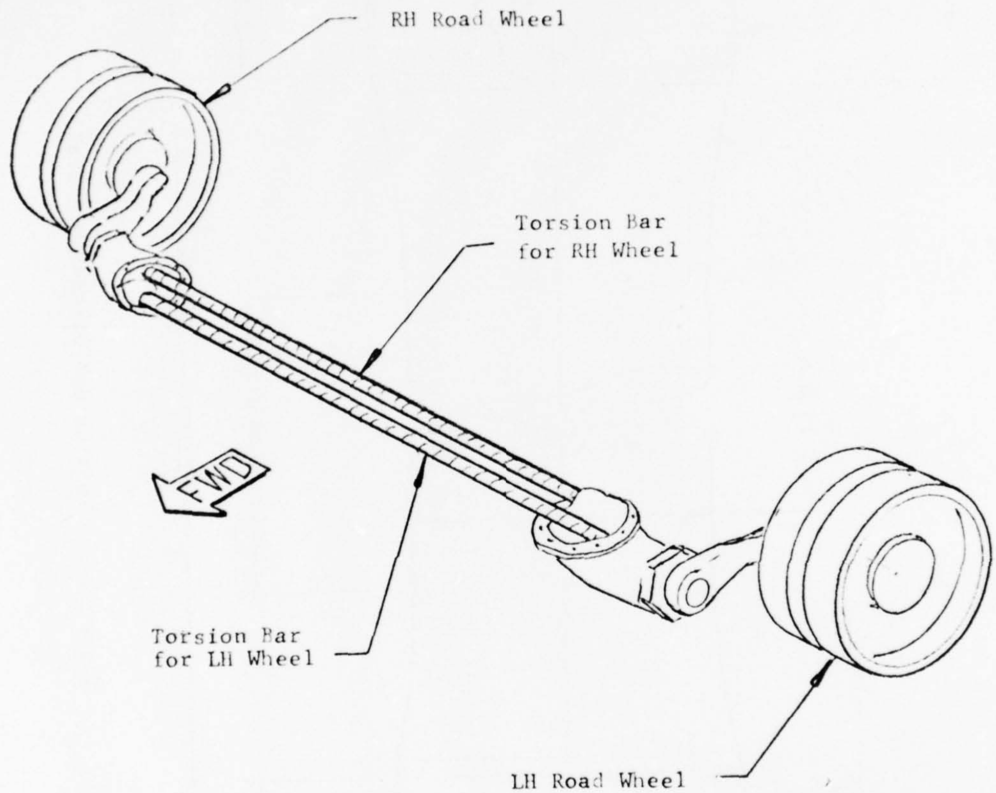


Figure 2-1 Torsion Bar Installation

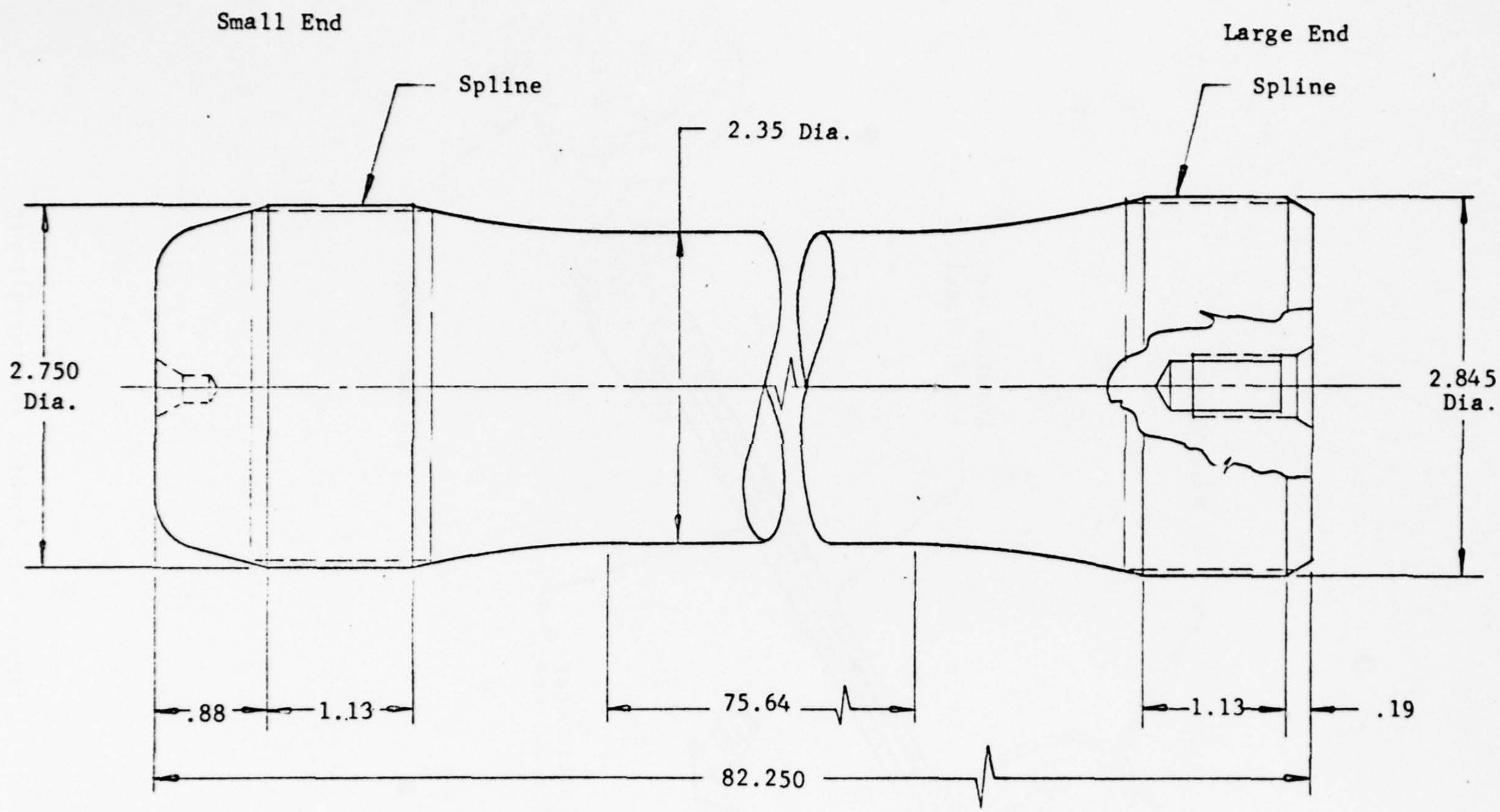


Figure 2-2 Torsion Bar Dimension (Metal)

The dynamic loads are 15 g. Applying the dynamic loads to the wheel, the wheel shock load may reach 15 x 11,250 = 168,750 lbs. Since the torsion bar at its maximum deflection balances a wheel load of only 23,135 lbs., it was assumed that a twist limitation is provided protecting the bar from angular deflections larger than 50.5°.

Per MIL-S-45387B, the spring shall have an endurance life of not less than 45,000 cycles. Consequently, the following spring diagram design requirements were assumed:

Spring Rate	7,330 in. lbs/degree
Minimum Torque	$4,075 \cdot 16 = 65,200$ lb. in.
Maximum Torque	$7,330 \cdot 50.5 = 370,165$ lb. in.
Maximum Cycles	50,000 (at maximum torque)
Diagram	Linear
Min/Max Stress Ratio	$\frac{65,200}{370,165} = .18$

The design load diagram for the torsion bar is shown in Figure 2-3.

2.2.1.4 Weight of the Steel Torsion Bar

The weight of the steel bar was calculated from the drawing per Figure 2-2. A material density of .238 lb/in.^{3} was assumed. Calculated weight is 103 lbs.

The weight given in Appendix D, Table 1 of the Specification and Requirements is 105 lbs.

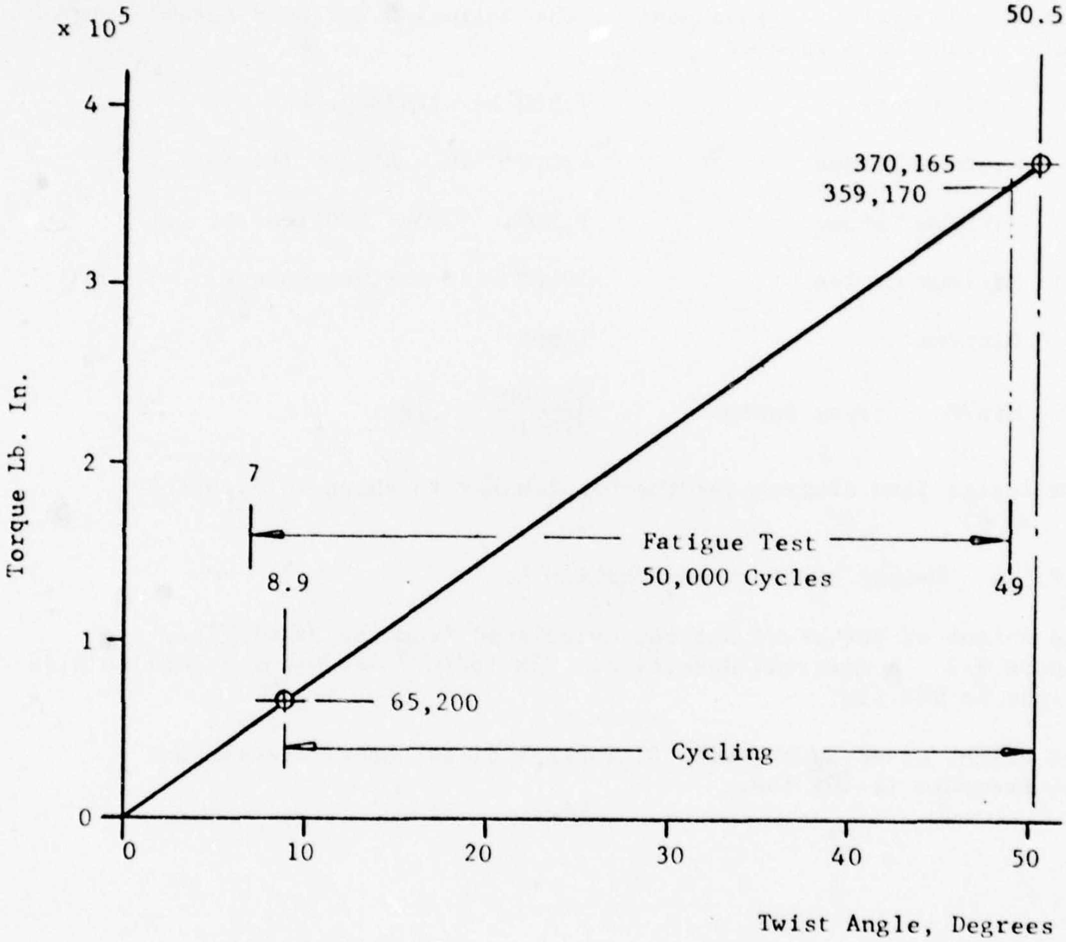


Figure 2-3 Load Diagram, Torsion Bar

2.2.2 Feasibility Study

2.2.2.1 Introduction

Experience has shown that to replace a torsion spring made of steel with one designed from fibrous composite materials it is difficult (or impossible) to meet both strength and stiffness requirements while at the same time staying within the same envelope.

The following analysis compares four different composite materials, and shows that for any of these materials an increase in the diameter or length of the torsion bar is necessary in order to meet both strength and stiffness requirements. An increase in the envelope is not acceptable, however, since the composite torsion bar would not be interchangeable with the present steel torsion bar and would involve redesign of other components.

2.2.2.2 Material Candidates

A composite material consisting of parallel fibers has non-uniform properties. Tension and compression strength and stiffness are high in the direction of the fiber. Shear strength and modulus depend on the matrix material and are low in a laminate with unidirectional fibers. The shear strength and modulus can be increased significally by orienting the fibers in adjacent plies 90° to each other and ±45° to the direction of the shear force.

Material properties of the composite also depend on the fiber content. A high fiber loading is desirable for strength and stiffness, however good shear properties and manufacturing considerations require an adequate resin content. For structural applications a fiber content of 55 to 65 $^{v}/_{o}$ is desired. Of further importance is the reduction of the allowable stress due to repeated loading (cycling). The magnitude of the reduction depends on the endurance properties of the fiber and resin and on the minimum to maximum stress ratio during cycling.

The principal fiber materials being considered are:

(1) Boron	AVCO, "Borsic" 5505/4
(2) Graphite	HTS
(3) Aramid	DuPont, "Kevlar" -49
(4) Glass	Owen-Corning, S2-449

Table 2-1 summarizes the properties for these materials in a composite laminate with 60 volume percent fiber and a thermosetting epoxy resin system as matrix material. The values were obtained from "Advanced Composite Design Guide" of the Air Force Systems Command, WPAFB, Ohio, from supplier's literature and from other information available at Riggs Engineering Corporation (REC).

The cycling shear stress allowables were estimated from Goodman diagrams for composite materials developed by REC. The percentage of ultimate stress allowed for 50,000 cycles differs somewhat for the different fiber materials. However, an average value of 57% was selected for all materials for the required R-factor of .18.

The diameter of the present steel torsion spring is 2.35 inches. It appears, however, that the maximum diameter that would fit the envelope is 2.70 inches. The diameter depends on stiffness requirements and on stress allowables. For a minimum diameter the allowable shear stress and the ratio F_{allow}/G must be a maximum value. For the candidate materials this ratio is:

Boron	44/7900	=	.00557
Graphite	38/5500	=	.00691
Aramid	16/3000	=	.00533
Glass	20/2200	=	.00909
Steel	140/11000	=	.01273

This comparison shows graphite composite to be the best candidate material. Boron has higher shear allowables, but its shear modulus is also very high resulting in a smaller ratio than graphite. Aramid and glass composites have low allowable shear stresses and thus a torsion spring made from these materials would require a larger diameter.

2.2.2.3 Sizing of the Composite Torsion Bar

For a preliminary analysis it was assumed that the torsion bar is manufactured from a single composite material wrapped around a mandrel of .50 in. diameter. The outer diameter of the composite bar was then calculated as follows:

For maximum allowable shear stress at the outer diameter (D = 2R):

$$f_s = \frac{2\, M_T\, R}{\pi\,(R^4 - r^4)}$$

Where r = inside diameter = .25 in. (constant).

Table 2-1

Composite Materials Properties

(Tape Laminate)

Fiber:	Unit	Boron	Graphite	Aramid	Glass
Filament Strength	10^3 PSI	500	400	525	665
Filament Modulus	10^6 PSI	58	40	19	12.6
Filament Density	lb/in^3	.094	.063	.052	.090
Composite:					
F_x^{tu}	10^3 PSI	192	180	200	226
F_x^{cu}	10^3 PSI	353	180	40	86
F_{xy}^{su}	10^3 PSI	15	12	9	7
F_{45}^{su}	10^3 PSI	77	66	28	35
F_{45}^{s} cycling	10^3 PSI	44	38	16	20
E_x^t	10^6 PSI	30	21	12	8
E_x^c	10^6 PSI	30	21	11	8
E_{45}^t	10^6 PSI		2.3		
G_{xy}^s	10^6 PSI	0.70	.65	0.30	.80
G_{45}^s	10^6 PSI	7.9	5.5	3.0	2.20
γ	lb/in^3	.073	.056	049	.071

Then R is determined from the equation:

$$R^4 - \frac{2\ M_T}{f_s\ \pi} R - r^4 = 0$$

and can be calculated by the trial and error method with the following results:

For boron, f_s = 44,000 psi:

$$R^4 - 5.197R - .00391 = 0$$

Then:

$$R_B = 1.7324 \text{ in. or } D_B = 3.465 \text{ in.}$$

For graphite, f_s = 38,000 psi:

$$R^4 - 6.017R - .00391 = 0$$

Then:

$$R_G = 1.8191 \text{ in and } D_G = 3.638 \text{ in.}$$

For the required stiffness, the twist angle corresponding to a torque moment of 359,170 in. lbs. must be 49° (see Figure 2-3). The twist angle is calculated from:

$$\theta = \frac{2\ M_T\ L_{eff.}}{\pi\ (R^4 - r^4)\ G} \quad [\text{rad}]$$

Where the effective length, $L_{eff.}$, is same as for the present steel torsion bar.

Then D is determined from the equation:

$$D = 2 \sqrt[4]{\frac{2\ M_T\ L_{eff.} + \theta\ \pi\ G\ r^4}{\theta\ \pi\ G}}$$

For boron, $G = 7.9 \cdot 10^6$ psi

$$D_B = 2\sqrt[4]{\frac{2 \cdot 359{,}170 \cdot 78.42 + .8852\,\pi \cdot 7.9 \cdot 10^6 \cdot .25^4}{.8852 \cdot \pi \cdot 7.9 \cdot 10^6}}$$

$$D_B = 2.534 \text{ in.}$$

For graphite, $G = 5.5 \cdot 10^6$ psi

$$D_G = 2.795 \text{ in.}$$

A summary of the results from the above calculations is shown below.

	Boron	Graphite
For Strength	3.465 in.	3.638 in.
For Stiffness	2.534 in.	2.795 in.

These results show that the boron spring would meet the envelope requirement if designed for stiffness. To meet the strength requirement the diameter would exceed the maximum 2.70 inches allowed. The graphite torsion bar will exceed the maximum diameter for both conditions.

2.2.2.4 Computer Analysis

A computer analysis was performed on the torsion bar in addition to the preliminary analysis in the previous paragraph for the purpose of optimizing the design.

For a torsion spring the most efficient section is a hollow shaft. However for increasing torque the outside diameter must increase with a corresponding decrease in wall thickness. Limitations on envelope often precludes an optimum diameter. Also as the wall thickness decreases, a point is reached, where torsional buckling occurs before the ultimate strength of the spring is reached.

Since the strain in the wall of a tube in torsion increases from the inner to the outer diameter, the material is only fully utilized at the outer diameter. A method to increase the efficiency in this respect is to use a higher modulus material towards the inner diameter

and lower modulus material towards the outer diameter. With graphite/epoxy composites it is possible to achieve this to some extent through the use of the various types of graphite fibers with a range of modulus (HTS, high strength with lower modulus, intermediate strength such as AS fiber, HMS, high modulus fiber or a combination graphite/steel).

The following analysis demonstrates a theoretical optimum tube based on a graded modulus concept. Although, in general, the strength of the composite will decrease with increasing modulus, for this analysis the strength will be assumed to be constant.

For comparison purposes a constant modulus torsion bar is also optimized.

2.2.2.5 Analysis of a Constant Stress Torsion Bar

The torque carried by a torsion bar can be calculated utilizing the following equation:

$$T = \int_{D_1}^{D_o} G\ \theta\ r^2\ (2\ \pi)\ (dr)$$

The model for this equation is shown on Figure 2-4.

The angle of twist, θ, per unit length is given by,

$$\theta = \frac{T}{\Sigma GI_p}$$

Where

ΣGI_p is the polar rotational stiffness and is calculated from

$$\Sigma GI_p = \int_{r_1}^{r_2} 2\ \pi\ r^3\ G(r)\ dr$$

Where the modulus will vary with the radius as follows,

$$G(r) = f(r)$$

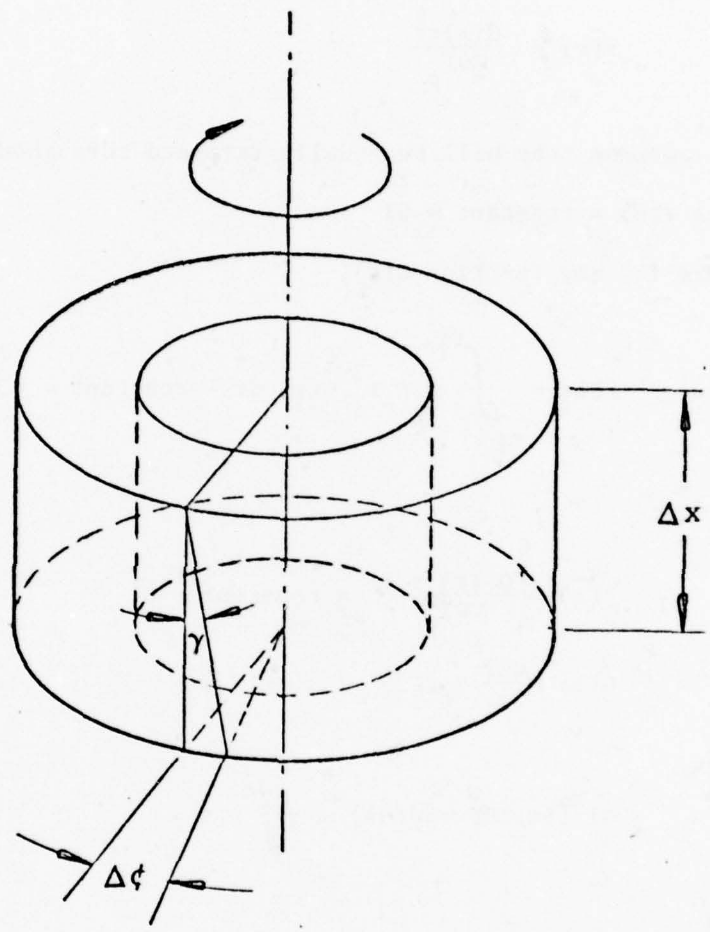


Figure 2-4 Torsional Deformation of Shaft Element

The shear stress in the shaft,

$$\tau = G\,\gamma = G\,r\,\theta$$

$$\tau(r) = \frac{G(r)rT}{\Sigma GI_p}$$

The optimum tube will be equally stressed throughout the wall thickness.

Thus $\tau(r)$ = constant = C1

Since for any function G(r),

$$\Sigma GI_p = \int_{r_1}^{r_2} 2\,\pi\,r^3\,G(r)\,dr = \text{constant} = C3$$

and

$$\tau(r) = \frac{G\,(r)\,r\,t}{\Sigma GI_p} = \text{constant} = C1$$

$$G(r) = \frac{C2}{r}$$

and

$$G1\ (\text{inside radius}) = \frac{C2}{r1}$$

and

$$G2\ (\text{outside radius}) = \frac{C2}{r2}$$

$$\int_{r_1}^{r_2} 2\,\pi\,r^3\,\frac{C2}{r}\,dr = C3 = \left.\frac{2\,\pi\,r^3\,C2}{3}\right|_{r_1}^{r_2}$$

$$C3 = \frac{2\,\pi}{3}\,C2\,\left(r_2^3 - r_1^3\right)$$

and

$$\tau = C1 = \frac{3T}{2\pi(r_2^3 - r_1^3)}$$

The spring constant of the torsion bar is K.

Angle of twist is,

$$\varphi = \frac{TL}{\Sigma GI_p}$$

$$K = \frac{T}{\varphi} = \frac{\Sigma GI_p}{L}$$

2.2.2.6 Torsional Buckling

As the diameter of the shaft increases the required wall thickness decreases. For thin walled tubes torsional buckling may become a problem. The basic torsional buckling equation for isotropic tubes of moderate length is,

$$\tau_{cr} = C1\ E\ t^{5/4} / (L^{1/2} \cdot r_2^{3/4})$$

With ends simply supported C1 = .7608

And with ends fixed C1 = 1.114

Moderate length tubular shaft is defined as,

$$10^2 < Z_t < 10^5$$

Where

$$Z_t = \frac{L^2}{r_2\ t}$$

For an orthotropic tube the equation becomes,

$$\tau_{cr} = C2 \ \frac{1}{2} \ \frac{E_x \ D_y^{5/3}}{L^{4/3} \ r_2^2}$$

Where

$$C2 = 3.6 \text{ for simply supported ends}$$

And

$$E_x = \Sigma \ E_{xi}^* \ t_o$$

$$D_y = \Sigma \ E_{yo}^* \ t_i^{\ 3}/12$$

For the hypothetical constant stress tube with $G(r) = C2/r$, and considering the layup to be all $\pm 45°$ plies, the values of E_x^* and E_y^* will be equal. Also the variation in E_x^* (or E_y^*) with fiber modulus is minor, i.e., approximately 2.0×10^6 psi for low modulus fiber and 2.4×10^6 for high modulus. Thus this variation will be neglected.

Based on $E_x^* = E_y^*$ the resulting τ_{cr} is identical to the isotropic case,

$$\tau_{cr} = C1 \ E^* \ t^{\ 5/4} \ / \ (L^{1/2} \cdot r_2^{\ 3/4})$$

2.2.2.7 Analysis of a Constant Modulus Torsion Tube

A more realistic approach from a manufacturing standpoint is a torsion tube with constant modulus (one type of material).

For this case the equations are;

For maximum stress in the tube,

$$\tau_s = \frac{T \ r_2}{G \ I_p}$$

Where

$$G\ I_p = \frac{G\ \pi\ (D_2^4 - D_1^4)}{32}$$

The spring constant is,

$$K = \frac{G\ I_p}{L}$$

And the critical buckling stress is,

$$\tau_{cr} = .7608\ E\ T^{5/4} \Big/ \left(L^{1/2} \cdot r_2^{3/4}\right)$$

2.2.2.8 Parametric Study

Consider a range of torsion bars with the following parameters.

Length L = 78 in.

Outside Radius r_2 = variable

Inside Radius r_1 = .25 in. (minimum practical)

Torque T = 370,000 in. lbs.

Maximum Shear Modulus G_{max} = 12 x 10^6 psi to 5.5 x 10^6 psi

Axial Modulus E^* = 2.2 x 10^6 psi

Spring Rate - 7330 in. lbs./degree twist

Maximum Angle of Twist = 50.5°

Allowable Shear Strength = Variable (Ult. 66,000 psi - Fatigue 38,000 psi)

Computer programs were developed for this study. Program ATS1 provides data for combinations of torsion bars with a variable modulus (constant stress) and program ATS2 provides data for constant modulus (variable stress).

The output data is shown on Tables 2-2 and 2-3.

```
ATS1   30-OCT-78   MU BASIC/RT-11 V01-01C

ACCEPTABLE TORSION BAR

I.D.= .5          O.D.= 4.4
SPRING CONSTANT= 429384       SHEAR STRESS= 16623.9
G1= 6.00000E+06               SHEAR BUCKL. STRESS= 241753

ACCEPTABLE TORSION BAR

I.D.= .5          O.D.= 4.2
SPRING CONSTANT= 435600       SHEAR STRESS= 19117.8
G1= 7.00000E+06               SHEAR BUCKL. STRESS= 234394

ACCEPTABLE TORSION BAR

I.D.= .5          O.D.= 4
SPRING CONSTANT= 429928       SHEAR STRESS= 22137.2
G1= 8.00000E+06               SHEAR BUCKL. STRESS= 226815

ACCEPTABLE TORSION BAR

I.D.= .5          O.D.= 3.8
SPRING CONSTANT= 414551       SHEAR STRESS= 25828.1
G1= 9.00000E+06               SHEAR BUCKL. STRESS= 218996

ACCEPTABLE TORSION BAR

I.D.= .5          O.D.= 3.6
SPRING CONSTANT= 430636       SHEAR STRESS= 30388.6
G1= 1.10000E+07               SHEAR BUCKL. STRESS= 210914

ACCEPTABLE TORSION BAR

I.D.= .7          O.D.= 3.6
SPRING CONSTANT= 436411       SHEAR STRESS= 30531.6
G1= 8.00000E+06               SHEAR BUCKL. STRESS= 194045

ACCEPTABLE TORSION BAR

I.D.= .7          O.D.= 3.4
SPRING CONSTANT= 413024       SHEAR STRESS= 36293
G1= 9.00000E+06               SHEAR BUCKL. STRESS= 185236

ACCEPTABLE TORSION BAR

I.D.= .7          O.D.= 3.2
SPRING CONSTANT= 420122       SHEAR STRESS= 43608.7
G1= 1.10000E+07               SHEAR BUCKL. STRESS= 176073
```

Table 2-2 Constant Stress Torsion Bar

```
ATS2    3-NOV-78  MU BASIC/RT-11 V01-01C

                    CONSTANT MODULUS BAR

ACCEPTABLE TORSION BAR

I.D.= .5        O.D.= 3.6
SPRING CONSTANT= 421901      SHEAR STRESS= 40476.2
G1= 2.00000E+06              SHEAR BUCKL. STRESS= 210914


ACCEPTABLE TORSION BAR

I.D.= .5        O.D.= 3
SPRING CONSTANT= 406763      SHEAR STRESS= 69970.8
G1= 4.00000E+06              SHEAR BUCKL. STRESS= 184806


ACCEPTABLE TORSION BAR

I.D.= .7        O.D.= 3.6
SPRING CONSTANT= 421454      SHEAR STRESS= 40519
G1= 2.00000E+06              SHEAR BUCKL. STRESS= 194045


ACCEPTABLE TORSION BAR

I.D.= .7        O.D.= 3
SPRING CONSTANT= 405871      SHEAR STRESS= 70124.7
G1= 4.00000E+06              SHEAR BUCKL. STRESS= 166514


ACCEPTABLE TORSION BAR

I.D.= .9        O.D.= 3.6
SPRING CONSTANT= 420409      SHEAR STRESS= 40619.8
G1= 2.00000E+06              SHEAR BUCKL. STRESS= 177463


ACCEPTABLE TORSION BAR

I.D.= .9        O.D.= 3
SPRING CONSTANT= 403780      SHEAR STRESS= 70487.8
G1= 4.00000E+06              SHEAR BUCKL. STRESS= 148616
```

Table 2-3 Constant Modulus Torsion Bar

Plotted on Figure 2-5 and 2-6 are the results of the computer studies.

The computer parametric study indicates for both the constant stress and the constant modulus torsion bar, that the stresses increase as the outside diameter decreases. As can be seen from Figure 2-5, the constant stress tube is limited to a minimum outside diameter as follows,

Outside Diameter in.	Allowable Angle of Twist	Required Allowable Shear Stress PSI
2.6	24°	90,000 (1)
2.9	22°	66,000 (2)
3.4	18°	38,000 (3)

For a constant modulus torsion tube the minimum diameter relations are (Figure 2-6),

Outside Diameter in.	Allowable Angle of Twist	Required Allowable Shear Stress PSI
2.9	22°	86,000 (1)
3.0	21°	66,000 (2)
3.7	17°	38,000 (3)

On Figures 2-5 and 2-6 are also plotted the allowable shear stresses for boron and graphite and the maximum allowable diameter of the torsion spring. It is clearly apparent that a torsion spring of fibrous composite materials which will meet strength and stiffness requirements will not fit within the same envelope as the present metal torsion spring.

The limitations of acceptable torsion springs are shown above to be allowable angle of twist which is less than one-half than required.

---

(1) Buckling Critical
(2) Ultimate shear strength for 45° plied high tensile strength graphite/epoxy.
(3) Fatigue allowable graphite/epoxy

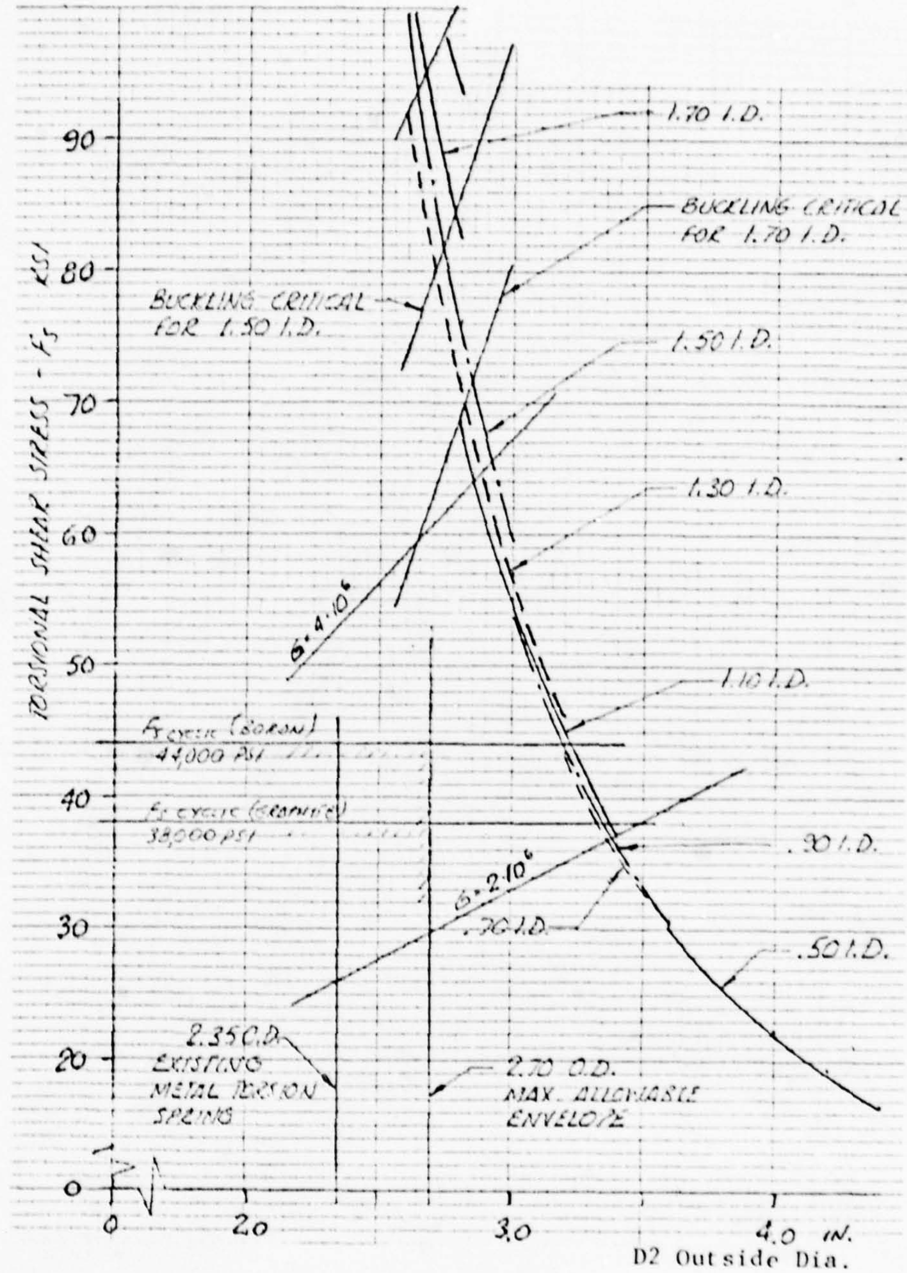


Figure 2-5 Constant Stress Torsion Bar

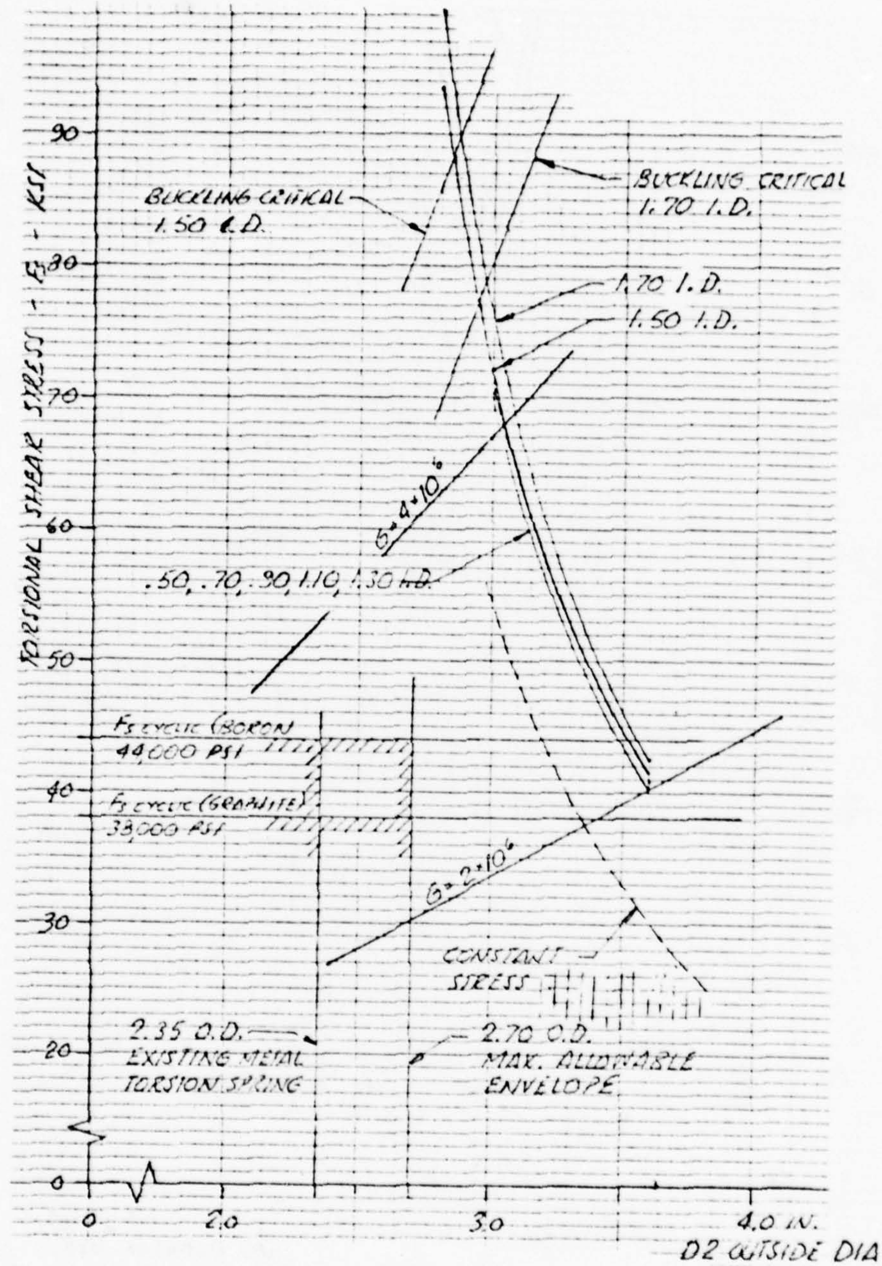


Figure 2-6 Constant Modulus Torsion Bar

2.2.2.9 Potential Use of Composite Materials for Torsion Bar

To further clarify the difficulty in replacing the present metal torsion spring with a composite spring the following analysis was made. It attempts to illustrate the range of stiffness parameters within which composite materials are suitable for torsion springs. This can be useful information for further designs.

The limitations of the material in a torsion spring is determined by the shear strain at the outer diameter of the tube or bar. Assuming that the stress-strain curve is linear within the allowable fatigue shear stress range we can write,

$$\gamma = \frac{F_s}{G}$$

Where

F_s = allowable shear stress

G = shear modulus

For a general thick walled tube,

$$\theta = TL/GJ \qquad (1)$$

or

$$T = \frac{G\theta}{L}\left[\frac{\pi}{2} \; R_o^4 \; (1 - R_i^4 / R_o^4)\right] \qquad (2)$$

$$R_i/R_o = 1 - t/R_o \qquad (3)$$

$$T = \frac{G\theta}{L} \; 2\pi \cdot R_o^3 \cdot t \; [1-1.5 \; t/R_o + (t/R_o)^2 - \frac{1}{4} \; (t/R_o)^3] \qquad (4)$$

$$K = T/\theta \qquad (5)$$

$$F_s = T \cdot R_o/J \qquad (6)$$

$$\gamma = T \cdot R_o/K \cdot L \qquad (7)$$

$$T = f_s \cdot \pi/2 \cdot R_o^3 \left[4 \cdot t/R_o - 6(t/R_o)^2 + 4(t/R_o)^3 - (t/R_o)^4 \right] \tag{8}$$

Solving for R_o and subtituting into Eq. (7),

$$\gamma = \frac{T^{4/3}}{K \cdot L} \sqrt[3]{\frac{1}{\pi/2 \cdot f_s}} \cdot \left[\frac{1}{\sqrt[3]{4 \cdot t/R_o - 6\,(t/R_o)^2 + 4(t/R_o)^3 - (t/R_o)^4}} \right] \tag{9}$$

For given values of $T^{4/3}/KL$ plots of f_s versus γ are shown on Figure 2-7. Also shown on Figure 2-7 are the shear stress strain curves for steel and for $\pm 45^\circ$ composites fabricated from boron, graphite, Kevlar, and glass fibers.

The stress strain relationship is shown linear in the stress range between zero and the allowable fatigue cycling stress.

For the present steel torsion bar the value of $T^{4/3}/KL$ is .807.

As can be seen from these curves, the potential for composite torsion bars occur when the loading stiffness parameter $T^{4/3}/KL$ is less than 0.30

1. Effect of Length - The stiffness parameter, $T^{4/3}/KL$, has only one design variable, length. By increasing length it would be possible to reduce this parameter and possibly utilize graphite/epoxy.

Shown on Figure 2-8 is the results of a length study versus weight. As can be seen, as the wall thickness of the tube decreases (t/R_o smaller), the weight decreases but the length requirement increases. For graphite/epoxy unreasonable lengths are required and are not practical. The buckling regime is also shown on Figure 2-8.

The conclusion remains that for the torsional moment, spring constant, and length required for the application, a composite torsion bar is not practical.

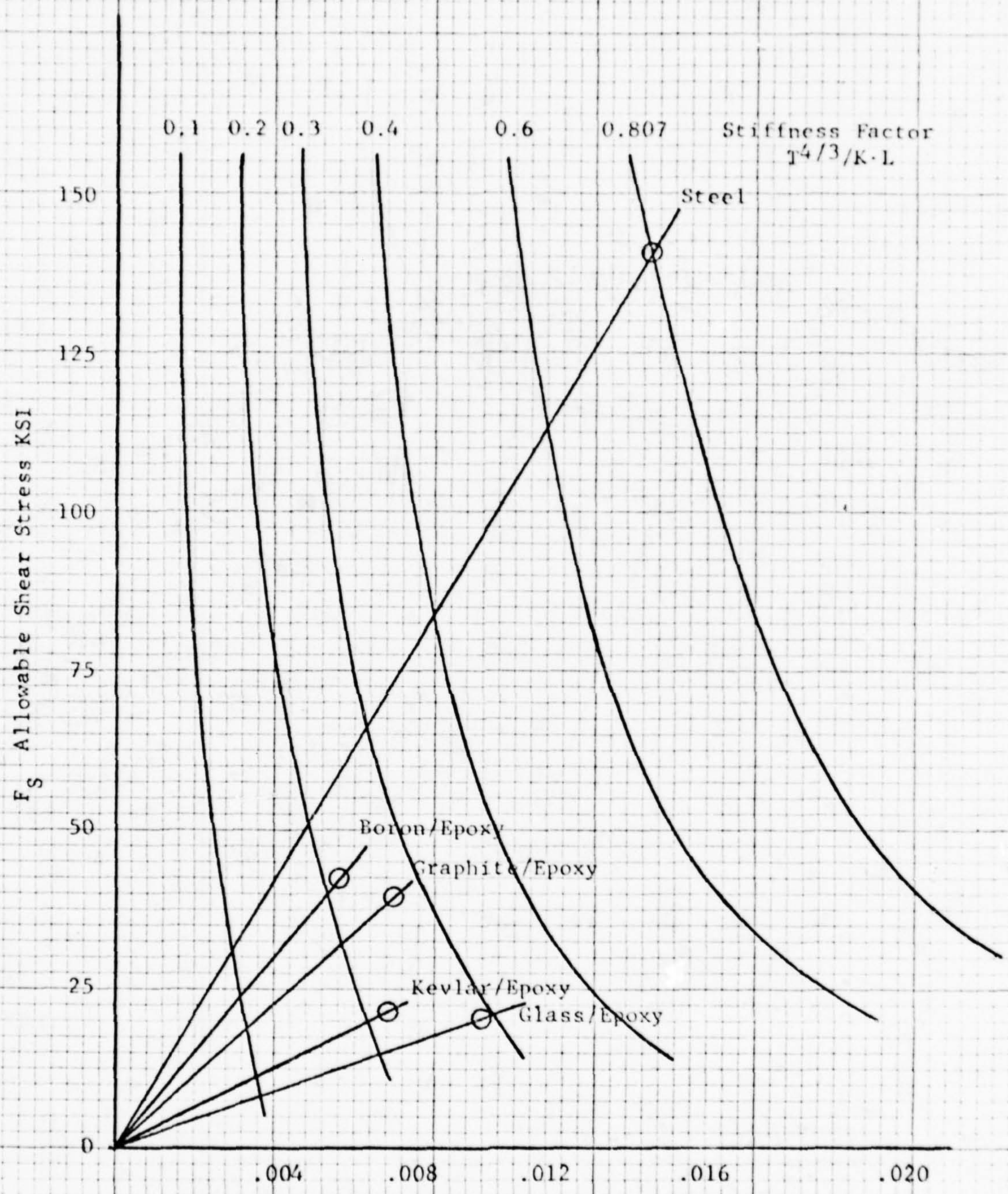


Figure 2-7 Shear Stress Strain Versus Stiffness Factor

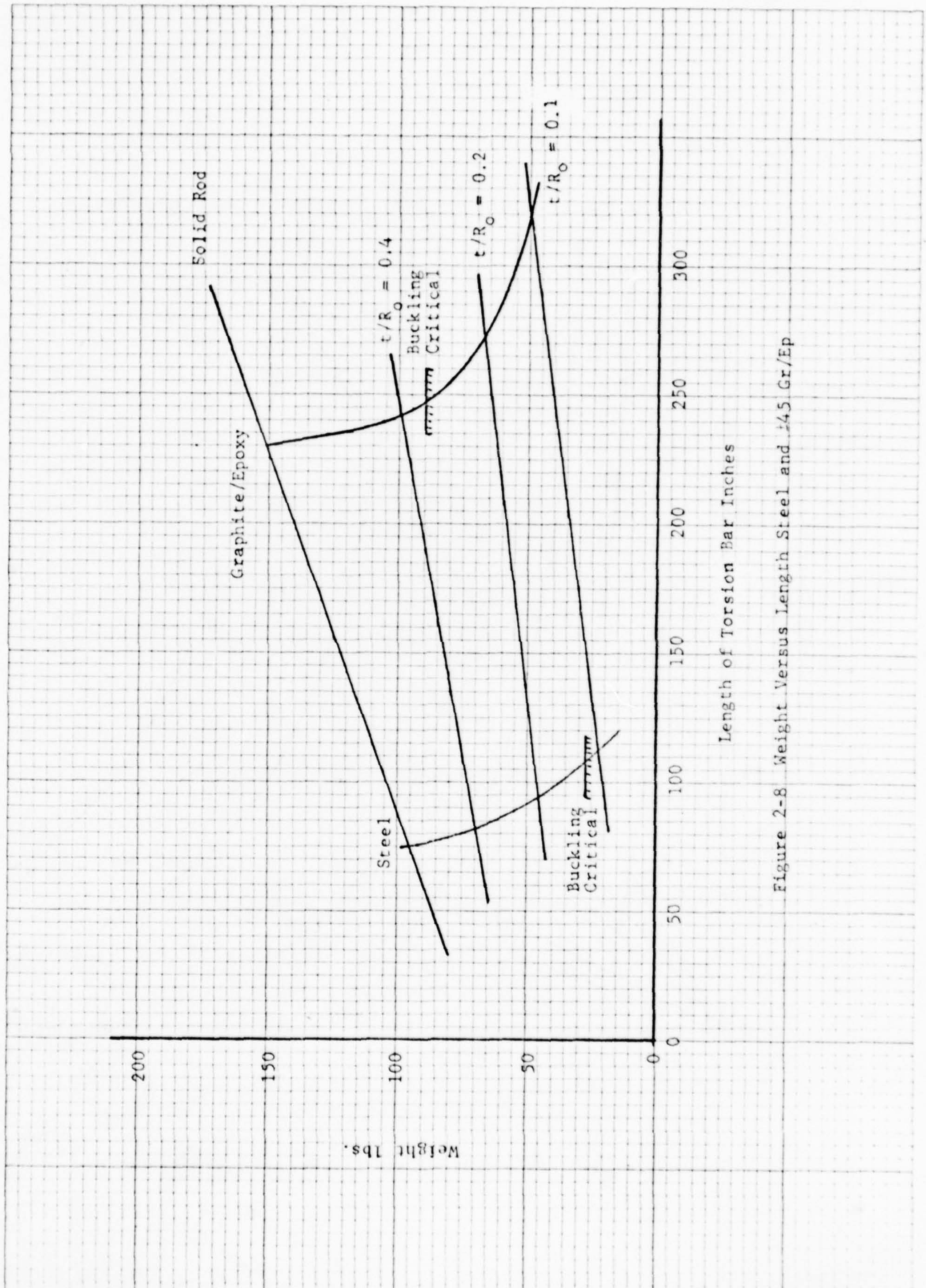


Figure 2-8 Weight Versus Length Steel and ±45 Gr/Ep

2.3 DRIVE WHEEL

2.3.1 Baseline Requirements

2.3.1.1 Description

The drive wheels transfer the engine torque to the tracks. There is one drive wheel for each track. The wheels are attached to the horizontal drive shaft located cross-wise at the rear end of the vehicle. Each drive wheel consists of a hub and two sprocket wheels. The assembly is bolted to the drive shaft flange. The teeth in the sprockets are shaped so that the connecting links, which tie the track shoes together form a continuous track, rest between the teeth. The drive torque is transferred at the points of contact between the connecting links and the sprocket teeth.

An exploded view of the drive wheel is shown in Figure 2-9

2.3.1.2 Dimensional Requirements

The baseline dimensional requirements were determined from drawings No. 7364134, Hub, Final Drive and No. 11637173 Sprocket. Drawing No. 11615320, End Connector was used to establish the contact area between the sprockets and connector link. The hub is attached to the drive shaft by an internal flange in the center of the hub. The dimensions for the flange diameter and the ten tapered bolt holes must remain as is.

The sprockets are bolted to flanges on each side of the hub. Due to severe wear of the sprocket teeths, the sprockets should be replaceable and separable from the hub. Therefore the dimensions for the external mounting flanges and for the eleven mounting holes must be maintained. The hub keeps the track centered by a guide groove. Due to severe wear, the surface of the guide groove should be made of steel.

Figure 2-10 shows the dimensional requirements for the hub.

The principal design dimensions for the sprocket are shown in Figure 2-11 which also includes the positioning of the end connector for reference. It should be noted, that there is an interference fit between the sprocket and the hub flange and that the mounting bolts are of a tapered design. This prevents any play between hub and sprocket under load. A wear limit line is indicated on two teeth of the sprocket.

The slightly conical hub which connects the center flange with the sprocket mounting flanges is assumed to have no tight dimensional restrictions. However the basic shape of the hub should be such that dirt is prevented from entering between the inner half of the hub and the driveshaft, or to accumulate inside the outer half of the hub.

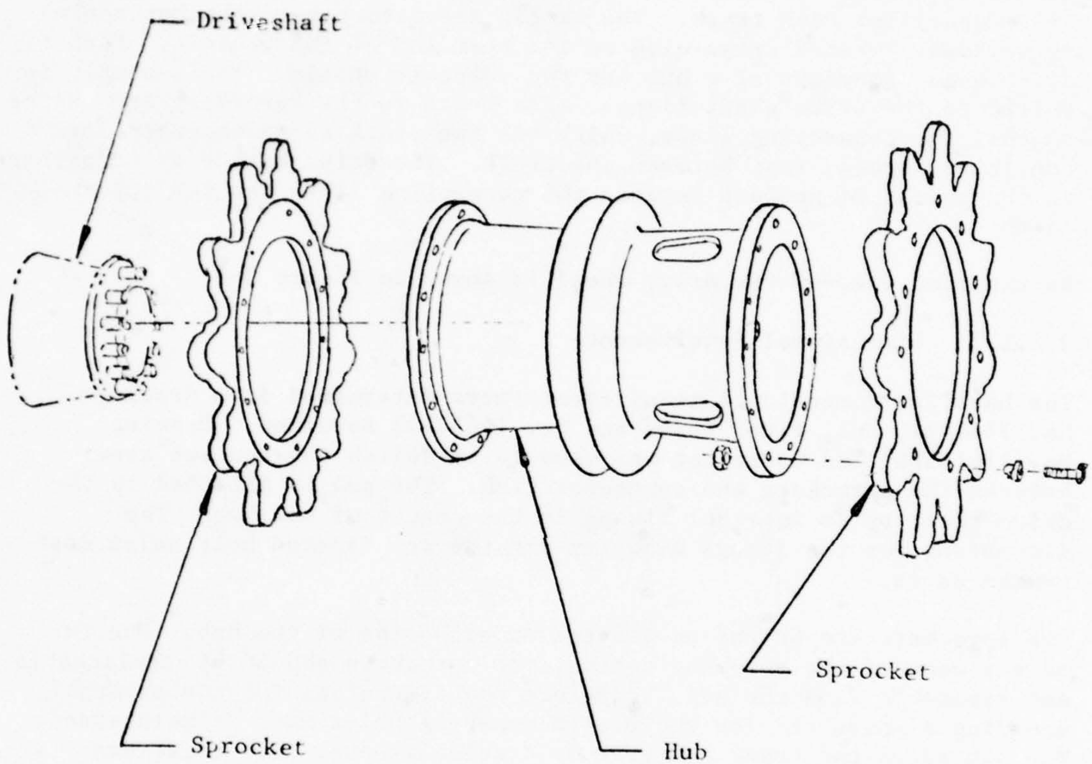


Figure 2-9 Drive Wheel and Sprockets (Metal)

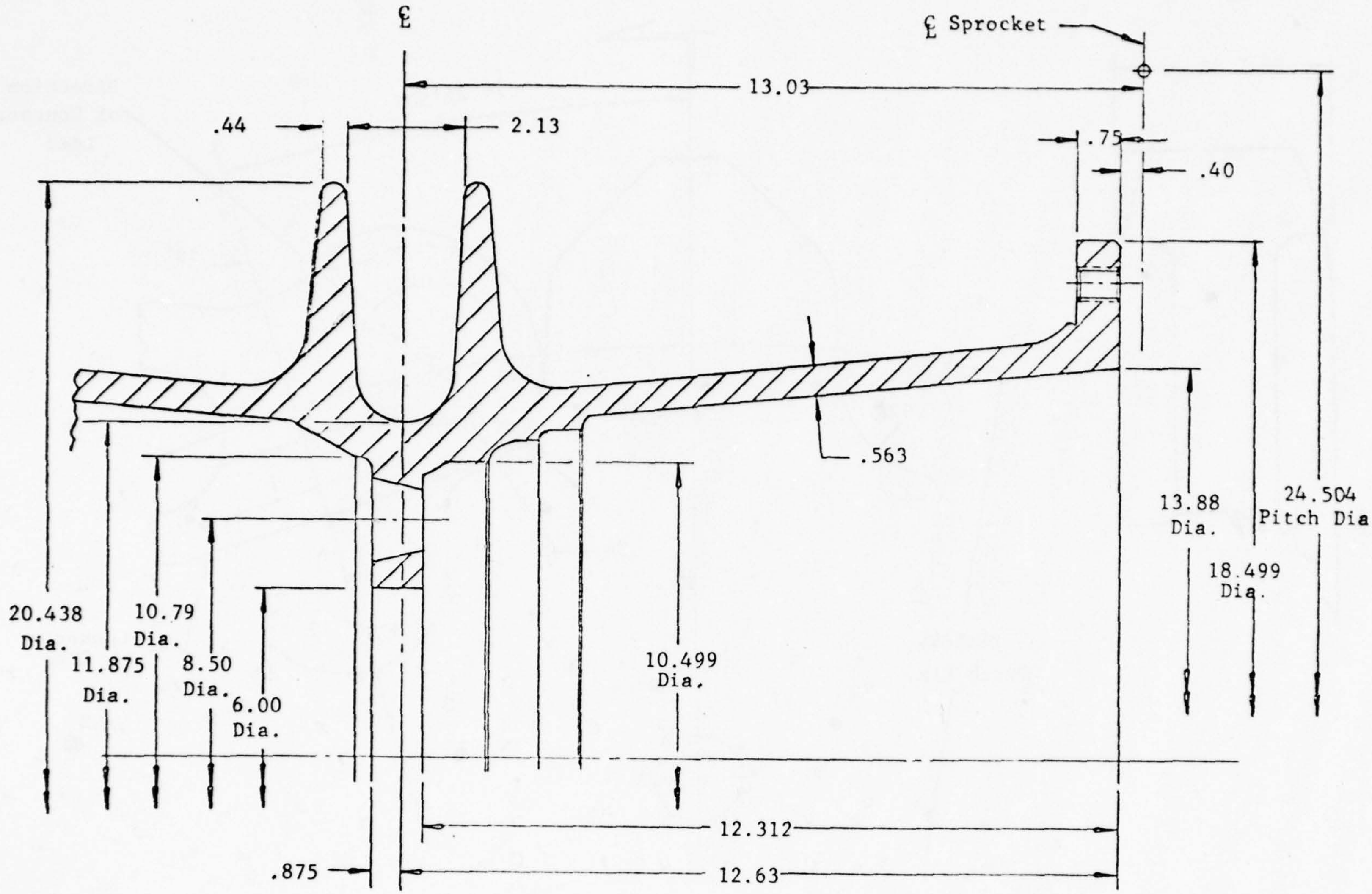


Figure 2-10 Dimensional Requirements of Drive Wheel Hub (Metal)

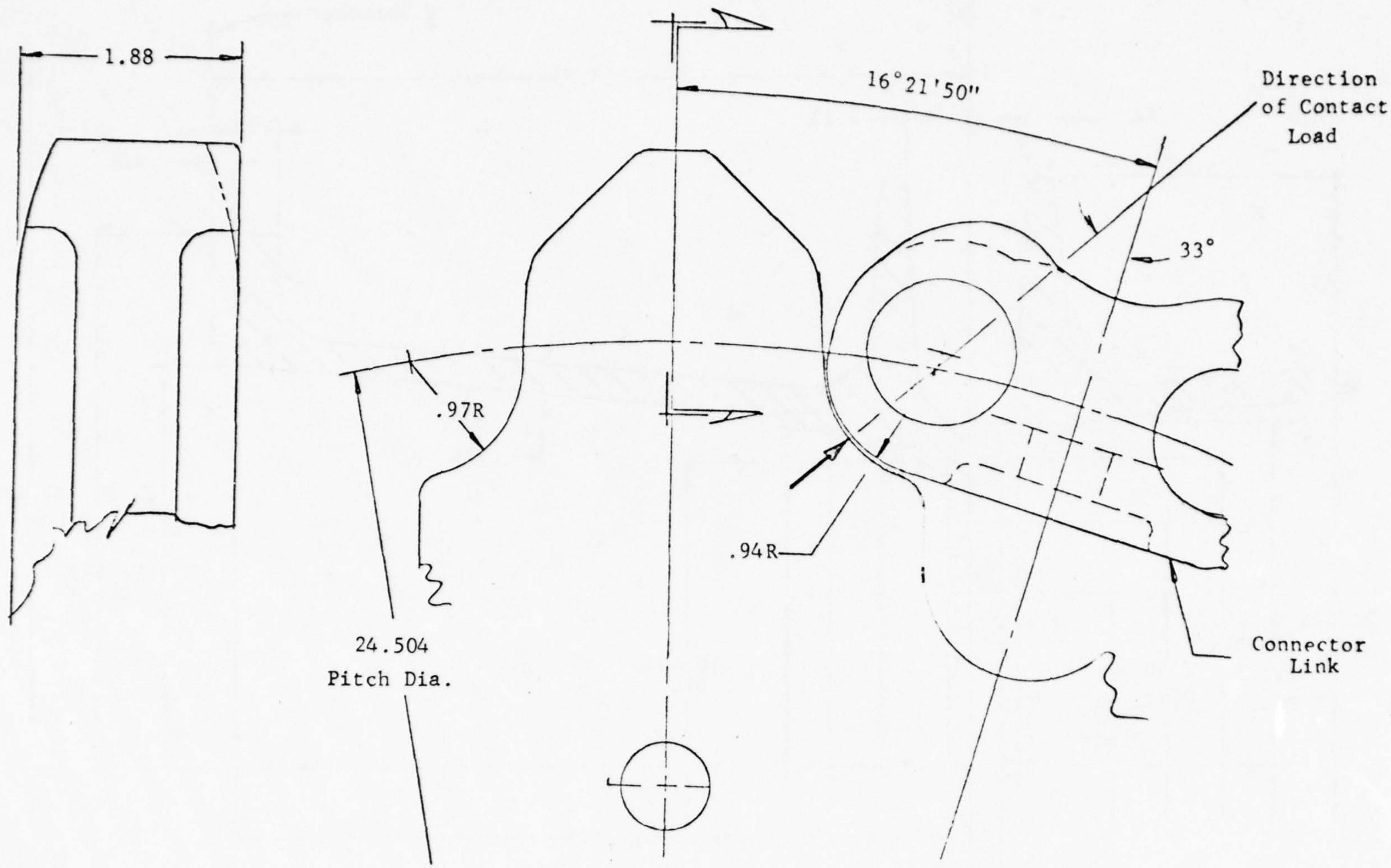


Figure 2-11 Sprocket Tooth and Connector Engagement

2.3.1.3 Design Load Conditions

The design load is specified in Appendix D, Table I, Item (2) of the Specifications and Requirements as follows:

Maximum Torque at Sprocket Pitch Line....80,000 ft-lbs.
(= 960,000 in.lbs.)

Maximum Track Tension....45,080 lbs.

The pitch diameter is specified on Drawing No. 11637173, P.D. = 24.504 in.

With the torque of 960,000 in. lbs., the track pull is 960,000 · 2/24.504 = 78,355 lbs. It is assumed, that each sprocket carries one half of the resultant load = .50 (45,080 + 78,355) = 61,717 lbs.

The contact angle between track and sprocket of 135° was determined by scaling Drawing No. 10905415. This angle varies somewhat with the deflection of the road wheel, but for the analysis it was assumed to be constant. It is equivalent to a four teeth engagement. Figure 2-12 shows the load diagram for a forward driving maximum torque condition.

The resultant maximum forces acting on the hub are accordingly:

Torsion (const.)	960,000 in. lbs.
Shear (const.)	79,274 lbs.
Bending Moment at Center	1,032,940 in. lbs.

The shear loads due to maximum torque at the bolt holes are:

at the driving shaft flange

$$P_{DF} = \frac{960,000}{4.25 \cdot 10} = 22,588 \text{ lbs/bolt}$$

at the sprocket flange

$$P_{SF} = \frac{480,000}{8.50 \cdot 11} = 5,134 \text{ lbs/bolt}$$

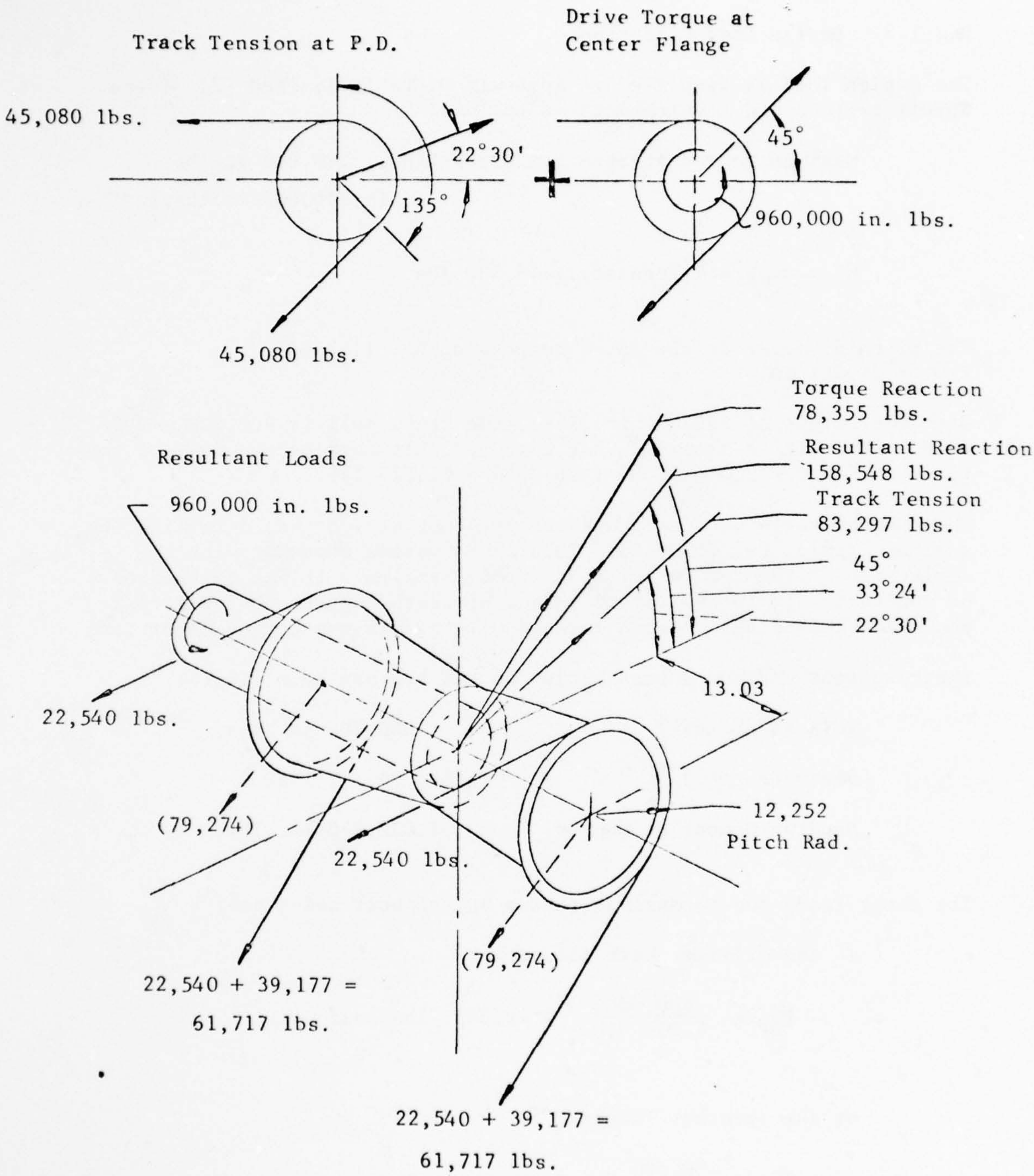


Figure 2-12 Design Loads for Hub

Some loads in the direction of the drive shaft axis, which are caused by the track plate, are also present, but are not specified.

The drive torque is transmitted to the track plate by the connector link and the engaged sprocket teeth. With only four teeth engaged the load per one tooth is in the tangetial direction.

$$P_T = \frac{480,000}{12.252 \cdot 4} = 9,794 \text{ lbs.}$$

assuming that all four teeth carry the torque load uniformly.

The actual load on the sprocket teeth depends much on the relative stiffness of the sprocket and of the track plate as well as on tolerances and wear. Therefore it is actually statically undetermined.

For an estimate of the sprocket tooth load due to track tension it was assumed, that the reaction load of 41,645 lbs. is distributed to the connector links parabolically as shown in Figure 2-13. By projecting these loads normal to the connector link it was found, that the maximum radial load acting on a link is 13,296 lbs. Since the contact pressure load between link and the tooth radii (See Figure 2-11)is inclined 33° to the center line of the connector link, the maximum contact pressure due to pre-tension is 7,927 lbs.

By converting the tangential drive force load on one tooth of 9,794 lbs. into the contact direction, a contact force of 17,983 lbs. is obtained. Consequently the maximum contact load between tooth and connector is 7,927 + 17,983 = 25,910 lbs. This load will be used for the design analysis of the tooth and the connector.

The sprocket is subjected to cyclic loading. The revolutions can be estimated from the vehicle velocity "V" and the pitch diameter "D" of the sprocket. Then,

$$n = \frac{V}{\pi D}$$

At a velocity of 30 mph and a pitch diameter of 24.504 in.,

$$n = \frac{30 \cdot 63,360}{\pi \cdot 60 \cdot 24.504} = 412 \text{ RPM}$$

The frequency of load change is n/60 = 6.87/sec. For 3,000 miles at a maximum speed of 30 mph, the life expectancy is 100 hours or,

$$6.87 \cdot 3,600 \cdot 100 = 2.5 \cdot 10^6 \text{ cycles}$$

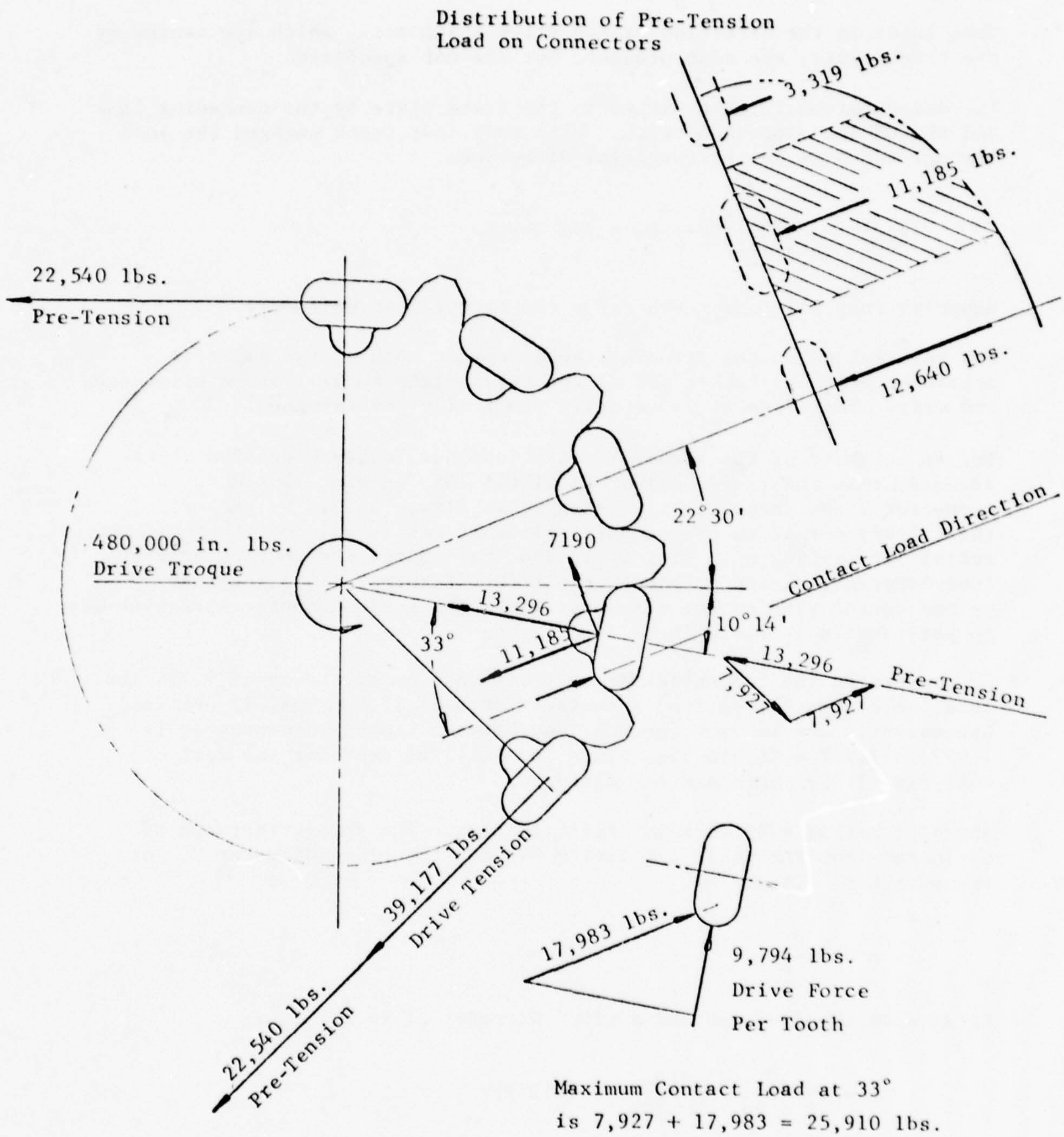


Figure 2-13 Sprocket Loads

2.3.1.4 Weight of Steel Drive Wheel

Per Appendix D, Table I, Item 2, of the Specifications and Requirements the weight of the drive wheel is:

Hub	303 lbs., density	$= .278$ lbs/in^3 (casting)
Sprocket	110 lbs., density	$= .283$ lbs/in^3

2.3.2 Feasibility

2.3.2.1 Introduction

Stresses in the present metal hub of the drive wheel were calculated to determine the design philosophy for the present design. It was found that the stresses were very low. For instance at the center of the hub the maximum bending stress in the cylindrical wall is 18,000 psi. Shear stresses are also very low.

The present hub is manufactured from a low alloy steel casting per QQ-S-681D with 90,000 psi ultimate tensile and 60,000 psi yield stress. The endurance limit in tension is 42,000 psi.

Comparison of these material allowables with the actual stress levels show that the safety margins are very high. It is therefore assumed that the hub design was based on stiffness rather than strength requirements. The same rules were adopted for the composite material design.

The stiffness parameters are expressed by:

E I for bending and

G J for torsional shear

Where

E = modulus of elasticity

I = area moment of inertia

G = shear modulus

J = polar moment of inertia

The stiffness parameters for the present metal hub at the center are:

$$E\,I = 29 \cdot 10^6 \cdot 334.5 = 9700 \cdot 10^6 \text{ lb. in.}^2$$

$$G\,J = 11 \cdot 10^6 \cdot 2 \cdot 334.5 = 7359 \cdot 10^6 \text{ lb. in.}^2$$

2.3.2.2 Material Selection

Properties for four composite materials are listed in Table 2-1 in paragraph 2.2.2. Because of stiffness requirements graphite and boron composites are better suited for this application than glass fiber or Kevlar. The boron fiber is expensive and difficult to handle in other than flat surface applications which leaves the graphite composite as the material best suitable for this application. The values for E and G are high as are the values for F_x and $F_{45°}$.

In addition to continuous fiber composites a moldable, short fiber composite material was consided for the sprocket. The material is designated Celcon GC-25A and is a thermosetting resin with 25% short glass fiber. It is manufactured by Celanese Plastics Company. Properties are listed under the discussion of the sprocket design Paragraph 2.3.2.6.

2.3.2.3 Hub Main Body

For the same stiffness for the center section of the metal hub the geometric parameters for a graphite composite hub should be:

$$I_{GR} = \frac{(EI)_{ST}}{E_{GR}} = \frac{9700 \cdot 10^6}{21 \cdot 10^6} = 462 \text{ in.}^4$$

$$J_{GR} = \frac{(GJ)_{ST}}{G_{GR}} = \frac{7359 \cdot 10^6}{5.5 \cdot 10^6} = 1338 \text{ in.}^4$$

E_{GR} and G_{GR} are obtained from Table 2-1.

In order to meet the requirements for both I and J, I must have a minimum value of 1338/2 = 669 in.^{4} since for the circular cross section of the hub, J = 2I.

The inside geometry around the internal flange area must remain unchanged because of mounting provisions to the drive shaft flange. The largest inside diameter is 10-11/16 or 10.69 in this area. The outside diameter is then:

$$D = \sqrt[4]{\frac{I + .049\, d^4}{.049}}$$

$$= \sqrt[4]{\frac{669 + .049\,(10.69)^4}{.049}}$$

$$= 12.784 \text{ in.}$$

This is considerably larger than the present design and would probably not be acceptable. The inside diameter of the track guide groove is 11.875 in. In inspecting one tank it appeared that there is some clearance between the bottom of the groove and the track guide, part No. 11599973, and that the diameter of the groove could probably be increased .25 in. without affecting the movement of the track guide.

For meeting the bending stiffness requirement the outer diameter should be,

$$D = \sqrt[4]{\frac{462 + .049\ (10.69)^4}{.049}}$$

$$= 12.246 \text{ in.}$$

The above numbers indicate that the stiffness in bending or torsion for a composite material hub cannot equal that of the present metal hub without changing the dimensions drastically.

Since maintaining the stiffness of the metal hub is not a specified requirement but ratheran assumed requirement, efforts were instead directed towards a design with the highest stiffness that could be achieved with composite materials while still maintaining the envelope of the present designs.

The main loads on the cylindrical portion of the hub is torsion and bending. The principal fiber orientations should therefore be $\pm 45^\circ$ and 0°. The ratio (fraction) of the fibers in each of these directions depends on the design requirements for strength and stiffness of the part.

The static material properties for 0° and $\pm 45^\circ$ orientations are known from technical literature, such as "Advanced Composites Design Guide". The composite properties for various ratios of $\pm 45^\circ$ and 0° fiber can be calculated from the known stress-strain diagrams of the basic orientations. For $2.5 \cdot 10^6$ cycles the allowable material properties are significantly reduced. Based on at REC available information for graphite/epoxy composites the allowable stress levels in percent of static ultimate stresses are:

$\pm 45^\circ$ in shear - 36%

0° in bending - 50%

The material allowables for high strength graphite are listed in Table 2-4 for a fiber volume of approximately 60%. The table includes the basic static properties for 0° and $\pm 45^\circ$ fiber orientation, hybrid composite laminates with various volume fractions of 0° and $\pm 45^\circ$ orientations, and the estimated design allowables at $2.5 \cdot 10^6$ cycles.

Table 2-4

Material Allowables for Hub Cone Design (Tape)

		Basic		Hybrid		
		0°	±45°	50/50	40/60	30/70
Static						
$F^{tu,\ cu}$	psi	180,000	23,200	97,500	81,000	6,450
F^{su}	psi	12,000	65,500	37,000	42,100	47,950
E	msi	21	2.34	11.7	9.8	7.9
G	msi	.65	5.52	3.1	3.6	4.0
At $2.5 \cdot 10^6$ Cycles						
$F^{t,c}$	psi	90,000	11,650	48,750	40,500	32,250
F^{s}	psi	4,320	23,580	13,320	15,150	17,260
E	msi	21	2.34	11.7	9.8	7.9
G	msi	.65	5.52	3.1	3.6	4.0

$\gamma = .056 \text{ lb/in}^3$

Due to the peculiar shape of the hub cone, the application of separate +45° and -45° plies may be inpractical and time consuming. From a manufacturing standpoint it is simplier to utilize woven graphite fabric which can be cut in form of segments and adjusted to the required cone configuration easier than tape material. The properties of the graphite fabric are summarized on Table 2-5 . Comparing the properties in Tables 2-4 and 2-5 it can be seen that the properties for 45° fabric (F^{su} and G) do not significantly deviate from the tape properties oriented at ±45°. However for resisting bending loads (F^{tu}, F^{cu}, E) the 0° oriented tape is more efficient than fabric.

Design problems to be overcome were mainly the following:

The critical section of the hub is at the center where the geometry of the internal flange and the track guide groove must remain as is and precludes any significant increase in wall thickness.

The torque moment must be transmitted from the internal flange in the middle of the double cone to the outer flanges at both ends of the hub by means of continuous fibers. The desired fiber orientation for this load is ±45°.

Bending load from both ends must be transmitted across the internal mounting flange by continuous fibers. The desired fiber orientation for this load is 0°.

Shear loads from the end bending loads must be transmitted to the internal mounting flange. The desired fiber orientation for this is ±45°.

Concentrated loads at the internal flange mounting bolts and at the sprocket mounting bolts must be resisted by the composite material without significant envelope modifications.

The flange for the track guide must be made of steel of adequate thickness to resist wear.

The composite hub design shall lend itself to state-of-the art manufacturing methods.

The composite hub shall be interchangeable with the present steel hub.

Table 2-5

Material Allowables for Hub Cone Design

(Graphite Fabric)

			Static	At $2.5 \cdot 10^6$ Cycles
0°				
	F^{tu}	psi	80,000	40,000
	F^{cu}	psi	88,000	44,000
	F^{su}	psi	19,000	6,840
	E^{t}	msi	10.3	10.3
	E^{c}	msi	8.5	8.5
	G	msi	1.04	1.04
±45°				
	F^{tu}	psi	27,300	13,650
	F^{cu}	psi	30,000	15,000
	F^{su}	psi	48,000	17,280
	E^{t}	msi	3.2	3.2
	E^{c}	msi	2.8	2.8
	G	msi	4.5	4.5

$\gamma = .057\ \mathrm{lb/in}^3$

After investigating a number of possible design approaches for the hub cone, two design concepts have been selected for evaluation and comparison.

Concept "A" - Integral Composite Design

Concept "B" - Composite Hybrid Design

Concept A is shown in Figure 2-14 and Concept B in Figure 2-15

2.3.2.4 Concept "A"

Torque and bending shear are transferred by $\pm 45^\circ$ oriented graphite fabric laminate of a nominal thickness of .45 in. This thickness is constant over the length of the cone. At the internal flange and at the sprocket mounting flanges the laminate is thicker due to inserted metal shims for increased bearing strength of the holes. To prevent galvanic corrosions the shims are made of 301 type stainless steel. The shims are .025 in. thick which makes the thickness of the internal flange .55 in. per cone. The two cones are not identical in length, because the mounting plane of the drive shaft, is offset 5/16 in. to one side of the guide flange centerline.

Bending and side loads are transferred by a laminate with essentially 0° oriented graphite fiber tape. The laminate follows the outer contour of the torsion cones, except at the central flange, where it bridges across from one cone to the outer. At this section it is 0.16 in. thick. The cross sectional area of the laminate is maintained constant between the flanges resulting in a slight variation in the wall thickness due to the change in the diameter.

Additional layers of $\pm 45^\circ$ oriented S2-glass fabric are placed outside the 0° oriented graphite laminate for additional torsional stiffness and for protection against external damage.

The track guide ring is designed as a one piece steel casting. The internal surface is machined smooth and provides the needed external support for the 0° laminated cone section. In-place molded rings of short fiber composites (bulk molding compound) provide lateral support for the guide flanges. Additional molded material is added to the inside of the cone for machining of the pilot hole for the drive shaft flange and for seal rings.

In order to make an approximate comparison of the stiffness in torsion and bending of the design concept with t he metal hub, the conical shape was replaced by a cylinder with a mean diameter of 12.750 in. The thickness of the $\pm 45^\circ$ graphite composite is .45 in., the thickness of the 0° graphite composite is .145 in. and the thickness of the $\pm 45^\circ$ glass fabric is .05 in. The total thickness is then .645 in. and the outer diameter 14.040 in. The shear modulus for glass fabric at $\pm 45^\circ$

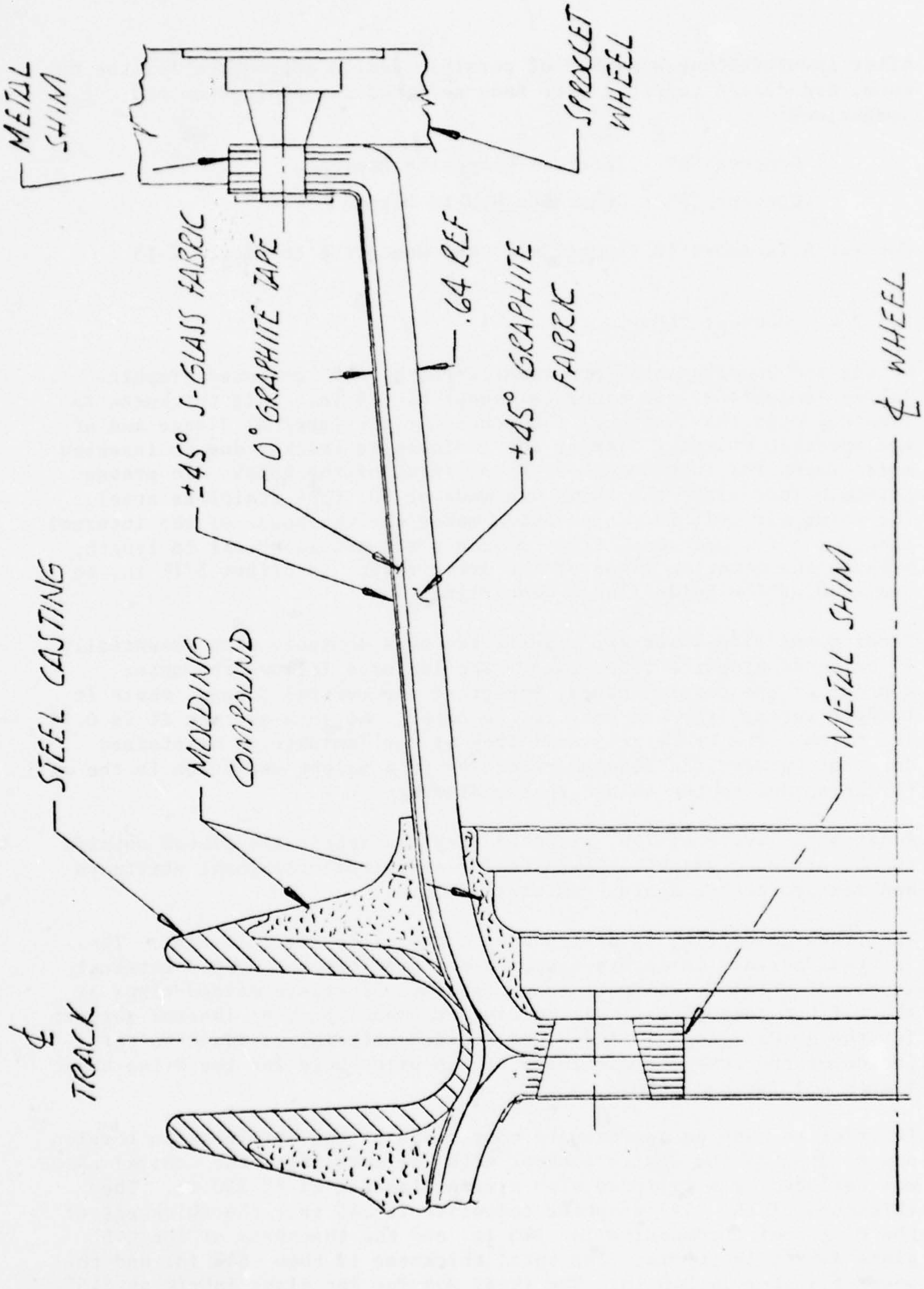


Figure 2-14 Drive Wheel Design Concept "A"

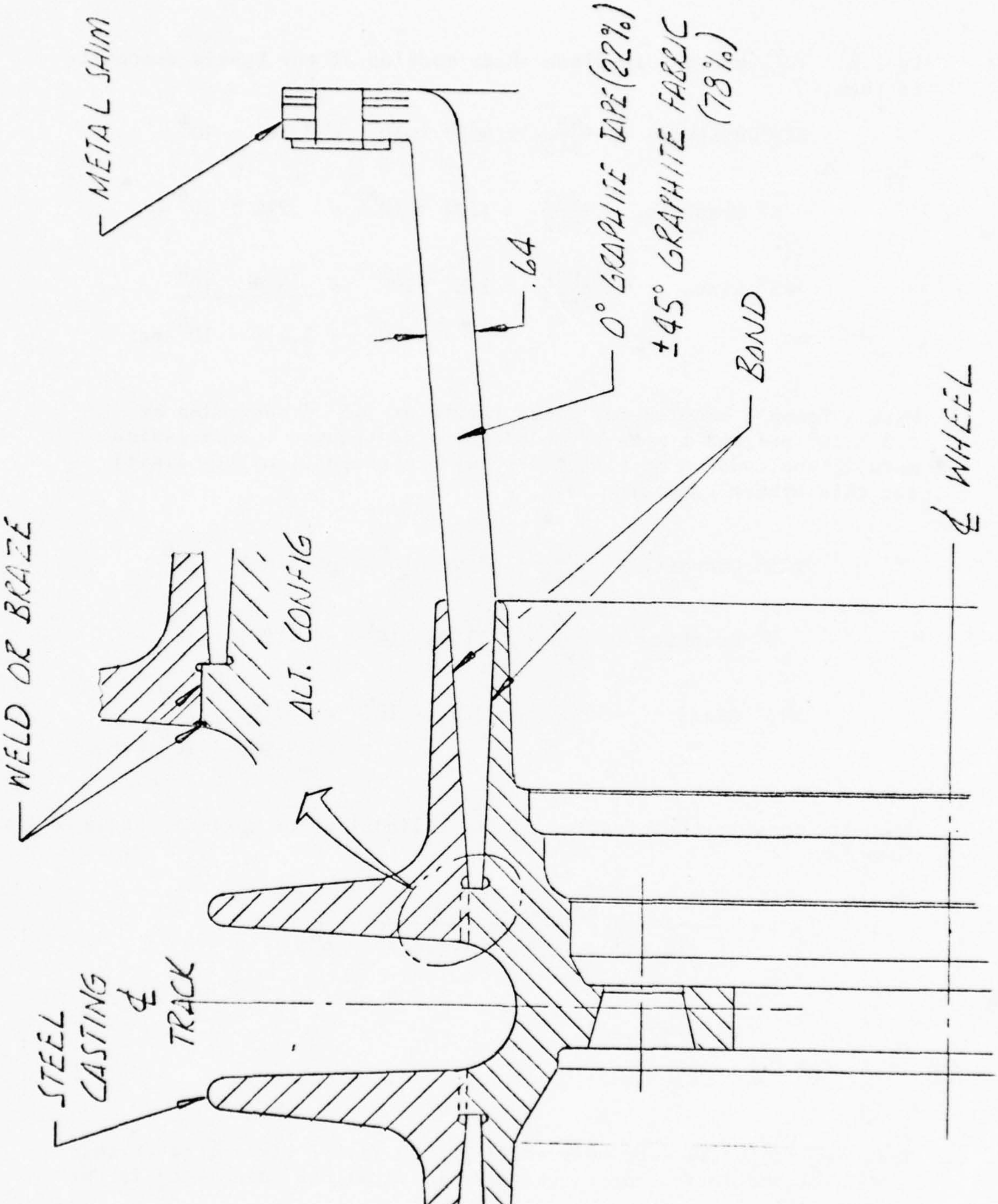


Figure 2-15 Drive Wheel Design Concept "B"

is $1.8 \cdot 10^6$, and the in-plane shear modulus of the hybrid composite is then,

$$\pm 45^\circ \text{ Graphite} - \frac{.45}{.645} \cdot 4.50 \cdot 10^6 = 3.141 \cdot 10^6$$

$$0^\circ \text{ Graphite} - \frac{.145}{.645} \cdot 1.04 \cdot 10^6 = .234 \cdot 10^6$$

$$\pm 45^\circ \text{ Glass} - \frac{.05}{.645} \cdot 1.8 \cdot 10^6 = \underline{.039 \cdot 10^6}$$

$$G = 3.414 \cdot 10^6 \text{ psi}$$

With a Young's modulus for glass fabric at ±45° orientation of $2.5 \cdot 10^6$ psi and a mean value of graphite tension - compression moduli, the modulus of elasticity (within proportionality limit) for this hybrid composite is,

$$\pm 45^\circ \text{ Graphite} - \frac{.45}{.645} \cdot 3.0 \cdot 10^6 = 2.094 \cdot 10^6$$

$$0^\circ \text{ Graphite} - \frac{.145}{.645} \cdot 21.0 \cdot 10^6 = 4.725 \cdot 10^6$$

$$\pm 45^\circ \text{ Glass} - \frac{.05}{.645} \cdot 2.5 \cdot 10^6 = \underline{.193 \cdot 10^6}$$

$$E = 7.012 \cdot 10^6 \text{ psi}$$

Then the torsional deformation of the cylinder under maximum torque load is,

$$\theta = \frac{32 \cdot 480{,}000 \cdot 12.63}{\pi\,(14.04^4 - 12.75^4) \cdot 3.414 \cdot 10^6}$$

$$= .001455 \text{ rad.}$$

$$= .0834^\circ = 5'$$

This deflection is extremely small but it is 2.8 times greater than the torsional deflection of the present metal hub calculated in the same manner.

The maximum deflection in bending for the composite cross section is calculated below,

$$A = \frac{\pi}{4}\ (14.04^2 - 12.75^2) = 27.14\ \text{in.}^2$$

and the composite moment of inertia,

$$I = \frac{\pi}{64}\ (14.04^4 - 12.75^4) = 609.09\ \text{in.}^4$$

$$y_{max} = 79{,}274 \left[\frac{12.63^3}{3 \cdot 7.012 \cdot 10^6 \cdot 609.09} + \right.$$

$$\left. + 2 \cdot \frac{12.63}{27.14 \cdot 3.414 \cdot 10^6} \right]$$

$$y_{max} = .0341\ \text{in.}$$

This deflection is also very small but it is three times greater than the deflection of the steel cylinder, used for comparison.

1. Bending Stress Analysis - Bending is mainly resisted by the axially oriented graphite fibers (0°). Since the E-Modulus of the ±45° oriented fabric layers is significantly less (about 14%) than that of the 0° oriented tape, the ±45° fibers do not contribute significantly, particularly at the center section, where the 0° composite bridges from one cone to the other.

At this section the dimensions of the 0° oriented fibers are:
I.D. = 11.50 in.   O.D. = 11.82 in.

The maximum bending stress is then:

$$f^b = \pm \frac{79{,}274 \cdot 12.63 \cdot 11.82}{.098\ (11.82^4 - 11.50^4)}$$

$$f^b = \pm\ 59{,}503\ \text{psi}$$

With an allowable fatigue stress of 90,000 psi, the margin of safety is,

$$M.S. = \frac{90,000}{59,503} - 1 = + .51$$

and for ultimate static stress,

$$M.S. = \frac{180,000}{59,503} - 1 = + 2.03$$

The actual stresses are slightly lower since the ±45° layers of graphite fabric and the ±45° layers of glass fabric do contribute a small amount of the strength of the hub.

2. Shear Stress Analysis - Shear stresses in the conical sections are caused by the drive torque and the bending loads. The drive torque generates a uniform shear flow in the wall of the cone, the bending load generates a variable (parabolic) shear flow. The maximum shear is the summation of the torque shear and the maximum bending shear.

The shear stress at the outer diameter due to torque is calculated from:

$$f^s = \frac{16\ T\ Do}{\pi\ (Do^4 - Di^4)}$$

and the shear due to bending load is given by:

$$f^s = \frac{Q \sin \alpha}{\pi\ R_{av}\ t}$$

where

T = Torque

Q = Bending load

α = Angle from load point

($\sin \alpha_{max} = 1$ at $\alpha = 90°$)

R_{ax} = Mean radius = $\frac{1}{4}$ (Do + Di)

t = Wall thickness = $\frac{1}{2}$ (Do - Di)

It is obvious, that the most critical section is the one with the smallest diameter, which is the section adjacent to the internal flange. At this point no assistance is provided by the 0° layers or the 45° glass outer layers. The selected critical section is on the outboard side of the internal flange with the section centroid approximately .60 in. from the center line of the hub. At this section the nominal inside diameter, Di = 10.6 in. and the outside diamter, Do = 10.60 + .90 = 11.50 in. R_{av} = 5.525 in. and t = .450 in.

Then the combined maximum shear stress in the ±45° graphite laminate is:

$$f^s_{max} = \frac{16 \cdot 480{,}000 \cdot 11.50}{\pi\,(11.50^4 - 1.60^4)} + \frac{79{,}274 \cdot 1}{\pi \cdot 5.525 \cdot .45}$$

$$= 5{,}778 + 10{,}099$$

$$= 15{,}877 \text{ psi}$$

Margin of safety for fatigue shear stress is,

$$M.S. = \frac{17{,}280}{15{,}877} - 1 = +.09$$

and for ultimate shear stress,

$$M.S. = \frac{48{,}000}{15{,}877} - 1 + 2.02$$

All other cross sections of the cone have larger diameters and the ±45° laminate is somewhat assisted in shear by the 0° composite and the ±45° glass layers. Consequently the stresses will be less than those calculated for the section close to the internal flange.

Tracing the load paths in the outboard and the inboard cones, see Figure 2-12, it becomes obvious that the two cones are not loaded symmetrically in the vicinity of the internal flange. In the outboard cone the torque shear as well as the bending shear is resisted by the mounting bolts on the drive shaft flange. In the inboard cone the torque shear is resisted by the mounting bolts but the bending shear is resisted by the bolts, and also to some degree by the drive shaft flange which has a tight fit in the hub. At this point the cone diameter

is already larger and the combined shear stresses smaller. The outboard flange does not have this support and is therefore more critical in shear, and was for this reason selected for the above shear stress analysis.

3. Analysis of the Internal Flange - As shown in Figure 2-14 the internal flange is made up of two flanges with each flange an integral part of the cone. For reasons described above the outboard cone is more severly loaded than the inboard cone. Therefore the stress analysis shall be made for the outboard flange only. Both flanges are of course bonded together and from an integral part of the hub. They are also bolted together to the drive shaft flange. This eliminates any possible bending deformation of the flanges and leaves only circumferential shear stresses and bolt bearing stresses to be considered.

The flange is designed with interspeced metal shims, for the purpose of increasing the allowable bolt bearing pressure, resisting shear deformation and distributing the load circumferentially within the flange laminate. The composite material in the flange laminate consists of the $\pm 45°$ oriented graphite fabric which is extended from the conical wall to form the flange.

The shims are .025 in. thin 1/4 hard stainless steel type 301 rings. This material was selected to avoid possible galvanic corrosion in contact with the graphite fibers. The mechanical properties of this material are:

$$F^{tu} = 125{,}000 \text{ psi}$$

$$F^{ty} = 75{,}000 \text{ psi}$$

$$F^{su} = 67{,}000 \text{ psi}$$

$$E = 27 \cdot 10^6 \text{ psi}$$

$$G = 12 \cdot 10^6 \text{ psi}$$

$$\gamma = .286 \text{ lb/in.}^3$$

The allowable bond shear strength is:

$$F^{su} = 2{,}500 \text{ psi}$$

$$F_{cycle} = 750 \text{ psi}$$

For an estimate of the required shim thickness and number, an Air Force Report was used. This reference report is called "Exploratory Application of Filament Wound Reinforced Plastic for Aircraft Landing Gear" AFML-TR-66-309, December 1966.

According to this report, the allowable shear load in a bolt hole is:

$$P = K_{br} \cdot A_{br} \cdot F_{tu}$$

Where

K_{br} = An experimental coefficient

A_{br} = Metal to metal contact area

F_{tu} = Ultimate strength of shim metal

The factor K_{br} depends on the shim thickness t_s and the bolt diameter D.

K_{br} was established as a function of D/t_s and it was found that for titanium shims the minimum shim thickness should not exceed D/t = 50. For other shim materials t_s can be estimated from:

$$t_s = \sqrt[3]{\frac{E_T}{E} \cdot t_{s_T}^3}$$

Subscript T = titanium

In addition the total shim bond area A_s shall be considered as well as the bond shear stress allowable F_s between shim and composite materials. Then,

$$P = F_s \cdot A_s$$

For the flange of the outboard cone, the average diameter of the tapered bolt hole is 1.47 in.

Then for titanium shim $t_{s_T} = \frac{1.47}{50} = .0294$ in. and for steel:

$$T_{S_{STL}} = \sqrt[3]{\frac{15}{27} \cdot .0294^3}$$

$$T_{S_{STL}} = .0242 \text{ in.}$$

The nearest .301 type sheet thickness available is .0250 in. or the equivalent titanium shim thickness = .0304 in.

Then $D/t_s = 1.47/.0304 = 48$. At this value $K_{br} = 1.71$ (Ref. AFML-TR-66-309, Figure 2-50). With four shims, the contact area is $A_{br} = 4 \cdot .025 \cdot 1.47 = .147$ in.^{2} and the allowable shear load per bolt:

$$P_1 = 1.71 \cdot .147 \cdot 125{,}000 = 31{,}421 \text{ lbs.}$$

The bond area of eight shim ring surfaces for one bolt is (average):

$$A_s = .7854 \left[(10.3^2 - 6^2) - 10 \cdot 1.47^2 \right] \frac{8}{10} = 30.46 \text{ in.}^2$$

With the allowable fatigue shear stress of $F_s = 750$ psi, the allowable bond shear load per bolt is,

$$P_2 = 30.46 \cdot 750 = 22{,}846 \text{ lbs.}$$

and for ultimate shear stress of 2500psi,

$$P_2 = 30.46 \cdot 2500 = 76{,}150 \text{ lbs.}$$

Without the shims and with an allowable cycling compression stress of 15,000 psi the allowable bolt load is,

$$P_3 = 15{,}000 \cdot 1.47 \cdot .45 = 9{,}923 \text{ lbs.}$$

The load on the mounting bolts consists of the torque load and the shear load. These loads are assumed to be evenly distributed over the ten mounting bolts which are located on an 8.5 in. diameter bolt circle.

The maximum load is at the bolt, where the tangential direction of the torque load and the shear load produce the largest resultant force. This is the case when the shear force is also tangential. The torque force per bolt is:

$$P_T = \frac{480{,}000 \cdot 2}{8.5 \cdot 10} = 11{,}294 \text{ lbs.}$$

And the shear force per bolt is:

$$P_S = \frac{79{,}274}{10} = 7{,}927 \text{ lbs.}$$

The maximum bolt load is thus 11,294 + 7,924 = 19,218 lbs. which shows that shims are necessary for bearing strength.

The margin of safety in bearing in the bolts is (based on P_2):

Cycling Stress

$$M.S. = \frac{22{,}846}{19{,}221} - 1 = + .19$$

Ultimate Stress

$$M.S. = \frac{76{,}150}{19{,}221} - 1 = + 2.96$$

The inboard half of the flange transmits mainly the torque load to the drive shaft bolts. Therefore the margin of safety there is higher.

Outside the area covered by the shim rings the full torsional and bending shear flow is present in the $\pm 45^\circ$ laminate. For the outboard flange, the critical shearing diameter is 10.3 in. and for the inboard flange 10.0 in. The nominal thickness of the composite plies is .45 in.

The resultant shear flow at 10.3 in. diameter is then:

$$f^s \cdot t = \frac{480{,}000 \cdot 2}{\pi \cdot 10.3^2} + \frac{79{,}274 \cdot 2}{\pi \cdot 10.3} \sin \alpha$$

The maximum shear stress due to bending occurs when $\sin\alpha = 1$.

$$f^s \cdot t = 2880 + 4900 = 7800 \text{ lb/in}$$

for a nominal thickness of .45 in. the maximum shear stress is then:

$$f^s = 7780/.45 = 17{,}289 \text{ psi}$$

The allowable stress for ±45° graphite fabric laminate for $2.5 \cdot 10^6$ cycles is 17,280 psi, which means that the margin of safety is zero.

To increase the margin of safety a doubler ring was added to the outboard flange. The ring is .045 in. thick stainless steel with an outside diameter of 11.25 in. Because the ring extends into the radius of the composite flange the outer edge must be formed into a radius. The doubler is bonded to the composite material.

For the inboard half of the internal flange the allowable bolt load with an average hole diameter of 1.20 in. is,

$$P_1 = 1.73 \cdot .12 \cdot 125{,}000 = 25{,}950 \text{ lbs.}$$

$$A_S = .7854 \left[(10^2 - 6^2) - 10 \cdot 1.20^2 \right] \frac{8}{10} = 31.16 \text{ in.}^2$$

$$P_2 = 31.16 \cdot 750 = 23{,}374 \text{ lbs.}$$

The applied force to the bolt is mainly the torque force, 11,294 lbs. No shear force on the bolt from the bending load is present, the bending shear being resisted by the tight fit between the drive shaft flange and the hub.

The shear stress in the composite material outside of the shims is,

$$f^s = \frac{480{,}000 \cdot 2}{\pi \cdot 10.3^2 \cdot .45} = 6{,}791 \text{ psi}$$

The margins of safety for the inboard flange are:

For bolt bearing

$$M.S. = \frac{23,374}{11,294} - 1 = +1.07$$

For shear in the composite material

$$M.S. = \frac{17,280}{6,791} - 1 = +1.54$$

Just inboard of the drive shaft flange, approximately 1 in. from the hub center full torque and bending shear forces are present in the composite laminate. The cone diameter at this point is however significantly larger than at the flange. The inside diameter D_i = 11.2 in., D_o = 12.1 in. and the nominal mean radius is 5,825 in. Following the same procedures used for analyzing the outboard cone the maximum shear stress is,

$$f^s = \frac{16 \cdot 480,000 \cdot 12.10}{\pi\ (12.10^4 - 11.20^4)} + \frac{79,274 \cdot 1}{\pi \cdot 5.825 \cdot .45}$$

$$f^s_{max} = 5,189 + 9,626$$

$$f^s_{max} = 14,815 \text{ psi}$$

Margin of safety for fatigue shear stress,

$$M.S. = \frac{17,280}{14,815} - 1 = +.17$$

For ultimate shear stress,

$$M.S. = \frac{48,000}{14,815} - 1 = +2.24$$

Between this point and the sprocket flanges the diameter of the cone increases. Theoretically the wall thickness could therefore be allowed to decrease in order to save weight, but it would also decrease the stiffness of the cone. From a manufacturing standpoint it is also desirable to maintain a constant material thickness in the cone.

4. Analysis of the Sprocket Flanges - The design of the sprocket mounting flange includes four thin metal shims in the form of rings or ring segments similar to the internal flange.

The flange fits tightly into the .53 in. deep recess in the sprocket. The close tolerance diameter is 18,500 in. nominal. The bending shear is resisted at this diameter and is not affecting the flange mounting bolts, which are loaded only by the torque moment of the sprocket of 480,000 in. lbs.

Assuming a uniform distribution of the torque load to the eleven bolts, which are located on a 17 in. diameter bolt circle, the maximum load per bolt is,

$$P = \frac{480,000 \cdot 2}{17 \cdot 11} = 5,134 \text{ lbs.}$$

The loads on the shim rings were calculated in the same manner as for the internal flange described earlier. The shims are made from .016 in. thick type 301 stainless steel (titanium equivalent thickness is .0196). With a bolt hole diameter of .65 in., the factor D/t_s becomes $.65/.0196 = 33$ and $K_{br} = 1.75$. Then $A_{br} = 4 \cdot .016 \cdot .65 = .0419 \text{ in.}^2$ and the allowable shear load per bolt:

$$P_1 = 1.75 \cdot .0419 \cdot 175,000 = 12,820 \text{ lbs.}$$

The bond area of all four shim rings is:

$$A_s' = .7854 \, [\, (17^2 - 15^2) - 11 \cdot .63^2 \,] \, \frac{8}{11} = 34.06 \text{ in.}^2$$

Then the allowable bond shear load:

$$P_2 = 34.06 \cdot 750 = 25,547 \text{ lbs.}$$

Without the shims the allowable bearing load is:

$$P_3 = 15,000 \cdot .65 \cdot .45 = 4,388 \text{ lbs.}$$

This is not sufficient and shows that metal shims are required. However the number of shim rings could be reduced and still maintaining a positive margin of safety. From a manufacturing standpoint it was decided that using the same number of shims as in the internal flange would be desirable. The margin of safety is then,

$$M.S. = \frac{12,820}{5,134} - 1 = +1.50$$

5. Analysis of Track Guide Flanges

The loads on the track guides are not known, however considering their function side loads are to be expected. These loads are transmitted from the guide on the track plate to the guide flanges on the sprocket by surface contact. Since the sprocket and the guide flange have the same angular velocity, no circumferential shear forces are introduced between the guide flanges and the sprocket hub. Due to operation requirements of the vehicle the environmental conditions are severe, and rocks may be carried by the fork like track guide and crushed as the track guide enters the flanged ring. Therefore it is necessary that the steel surface of the present guide flanges are maintained and that the lateral strength of the flange should be similar to the present steel flange. The design concept is shown in Figure 2-14.

The u-shaped ring is fabricated as a casting or weldment. Stiffness in the lateral directions is provided by fillet rings molded in place from short fiber (glass) molding compound. Because of the shape of the hub the steel ring with the molded rings must be inserted in the lay up tool prior to laying up the 0° (axial) oriented graphite fibers.

A comparative analysis was conducted with the strength and stiffness of the present steel flange as reference. A one inch wide segment was selected from the metal flange. With known ultimate material properties a hypothetical side load was determined at which the flange segment would fail in bending. The side load was applied to a one inch wide segment of the steel composite hybrid flange and the material stresses calculated. To compare the relative stiffness of the flange segment in bending, the EI of the steel and of the hubrid design were compared. Since this was a comparative analysis, no circumferential stresses in the flange were considered. The selected critical section is 2.75 in. from the outer diameter of the flange and the load was applied .25 in. from the outer diameter. The material properties are:

$$F^{tu} = 90,000 \text{ psi}$$

$$E = 29 \cdot 10^6 \text{ psi}$$

Composite molding compound, type Fiberite E260H:

$$F^{cu} = 30{,}000 \text{ psi} \qquad (F_{tu} \text{ value is higher})$$

$$E = 3.1 \cdot 10^6 \text{ psi}$$

At the critical cross section the present steel flange is .80 in. thick.

Then the critical bending load at the flange tip is:

$$P = \frac{90{,}000 \cdot 1 \cdot .80^2}{6(2.75 - .25)}$$

$$P = 3{,}840 \text{ lbs.}$$

and the bending moment, $M_b = 9{,}600$ in. lbs.

For the analysis of the hybrid reference section, the thickness of the steel flange is .44 in. and of the molding .80 in. To determine the effective modulus of inertia and the stresses, the width of the molded material was converted to an equivalent steel width which is the ratio of the moduli, or 3.1/29 = .107.

The effective modulus of inertia was then calculated, $I = .039$ in.^{4}, the section modulus for steel $Z_{stl} = .1221$ in.^{3} and for the molding $Z_m = .0427/.107 = .3987$ in.^{3}.

Then the maximum bending stress in the steel flange is,

$$f_{stl} = \frac{9{,}600}{.1221} = 78{,}624 \text{ psi}$$

and in the molded ring,

$$f_m = \frac{9{,}600}{.3987} = 24{,}078 \text{ psi}$$

which is within the allowable compressive stress for the molding material.

The stiffness for the two materials were compared,

Steel:

$$EI = 29 \cdot 10^6 \cdot 1 \cdot .80^3/12 = 1.237 \cdot 10^6 \text{ lb. in.}^2$$

Hybrid:

$$EI = 29 \cdot 10^6 \cdot .039 = 1.131 \cdot 10^6 \text{ lb. in.}^2$$

This comparison shows that the hybrid composite/steel guide flange might deflect slightly more than the present steel flange but would resist a higher side load.

6. Weight Analysis of Concept A

The calculated weight for the composite hub design concept A shown in Figure 2-14 is 113.3 lbs. Of that weight 57.6 lbs. or 51% is fibrous composite materials and 55.7 lbs. or 49% is steel.

The weight of the present steel hub is 303 lbs. Weight savings is then 303 - 113.3 = 189.7 lbs. or 63%.

2.3.2.5 Analysis of Concept "B"

The critical part of this concept is the design of the joint between the center steel section and the composite outer cone sections. (See Figure 2-15).

The dimensions of the internal flange, the track guide flanges and the drive shaft seat and seal are identical to the present all-steel design. No stress analysis of this section is therefore needed.

The composite cones were designed for the same stiffness as the cones of the all composite design, Concept A. The laminate consists of $\pm 45^\circ$ oriented fabric plies and 0° oriented tape plies similar to Concept A, but since the 0° plies are not continuous across the center section, they can be laminated interspersed with the 45° fabric plies. As a result, the composite cones can be fabricated separately and then joined to the center section. The two end cones including the sprocket flanges are identical.

The composite cone consists of 78% ±45° graphite fabric and 22% 0° graphite tape. Material properties were extrapolated from values listed in Tables 2-2 and 2-3 moduli:

$$E = 6.96 \cdot 10^6 \text{ psi}$$

$$G = 3.65 \cdot 10^6 \text{ psi}$$

Design allowables for $2.5 \cdot 10^6$ cycles,

$$F^{tc} = 277{,}000 \text{ psi}$$

$$F^{s} = 13{,}138 \text{ psi}$$

The stiffness of the cones were compared in the same manner as for Concept A above. A section with a mean inside diameter of 12.750 in. was selected. The wall thickness is .64 in. and the outer diameter 14.03 in.

Then the torsional stiffness parameter is expressed:

$$GJ = 3.65 \cdot 10^6 \cdot .098\ (14.03^4 - 12.75^4)$$

$$GJ = 4407 \cdot 10^6 \text{ lb. in.}^2$$

The equivalent stiffness parameter for the steel hub is $GJ = 7359 \cdot 10^6$ lb. in.^{2} indicating that the torsional deflection is 7359/4407 = 1.67 greater than for the steel hub.

The bending stiffness parameter is:

$$EI = 6.96 \cdot 10^6 \cdot .049\ (14.03^4 - 12.75^4)$$

$$EI = 4{,}202 \cdot 10^6 \text{ lb. in.}^2$$

Compared with the bending stiffness parameter for the steel hub of $9{,}700 \cdot 10^6$ lb. in.^{2} the composite section would deflect 9700/4203 = 2.3 times more in bending.

Both torsional and bending deflections for Concept "F" are thus somewhat smaller than for Concept "A".

1. Stresses in the Composite Cone Section - The maximum bending stress in the composite laminate occurs at the end of the tapered steel flanges. The inside cone diameter is 12.65 in., thickness .64 in. and the outer diameter is 13.93 in. The section modulus is:

$$Z = .098 \frac{13.93^4 - 12.65^4}{13.93} = 84.75 \text{ in.}^3$$

The bending moment at this section is 79,274 · 7.03 = 557,296 lb. in. The maximum bending stress is,

$$f^b = \frac{557,296}{84.75} = 6,576 \text{ psi}$$

and

$$M.S. = \frac{27,700}{6,576} - 1 = +3.2$$

The shear in this section is composed of torsional shear and bending shears.

The maximum shear stress is,

$$f^s = \frac{16 \cdot 480,000 \cdot 13.93}{3.1416\ (13.93^4 - 12.65^4)} + \frac{79,274 \cdot 1}{3.1416 \cdot 6.645 \cdot .64}$$

$$f^s_{max} = 2,827 + 5,933$$

$$f^s_{max} = 8,760 \text{ psi}$$

and

$$M.S. = \frac{13,138}{8,760} - 1 = +.50$$

for

$2.5 \cdot 10^6$ cycles

This analysis shows that the stresses in the critical composite hub section are low. A reduction of the wall thickness would be possible, but is not recommended due to associated increase of torsional and flexural deflections.

2. Analysis of the Bonded Joint - For weight considerations the center steel section should be as short as possible. The actual dimensions depend on the required length of the joint between the steel and the composite parts. The joint could be designed as a bolted or riveted joint, as a bonded joint, or as a combined bonded-bolted joint.

The main problem to be resolved is to develop a joint concept whidh would resist bending and shear loads for $2.5 \cdot 10^6$ fully reversed ($R = -1.0$) cycles. Under such load conditions the reduction of the static design allowables is significant. In particular a bolted or riveted joint is not very efficient in cases involving reversed loads. A bonded joint has usually a better fatigue life for $R = -1.0$ conditions. A combination of bonded and bolted design is difficult to analyze realistically without getting into problems of estimating minute deformations in bondline, bolts and adherends.

For this feasibility study it was therefore decided to base the design of the Concept "B" hub on a pure bonded joint.

Analysis has shown that a double lap joint of 4 in. length is required. Since it is difficult to machine a tapered groove of this depth in the steel casting, it is necessary to machine the two concentric flanges separately and then assemble them by welding or by threading as is shown in Figure 2-15.

The stress analysis of the bonded joint was carried out based on information and design procedures published in AFML's "Advanced Composites Design Guide" Volume I and II.

3. Establishing of Design Allowables for the Bondline - The shear stress allowables depend on:

Materials of adherends

Fiber orientation of the adherend composite

Thickness of the adherends

Adhesive system

Thickness of the adhesive in the joint

Load condition

Cycle number

Environment

Fabrication quality

The "Design Guide" does not contain information for a joint which is exactly the same as the one being analyzed and some assumptions were necessary.

The Design Guide lists the following design allowables for cycle tests on boron-titanium joint with $L/t = 8$ (overlap length/adherend thickness) made with two different adhesive systems.

Adhesive	AF126-2	HT424
F^s static, psi	3990	2432
F^s $2.5 \cdot 10^6$ cycles at $R = -1.0$ % of static	10	27
F^s cycle allow. psi	399	657

The material properties of the two adhesives are (tested with aluminum adherends):

Adhesive	AF126-2	HT424
G psi	89,000	309,000
F^{su} psi	5,220	4,500
γ^{ult} in/in	.80	.075

This shows, that the somewhat more rigid adhesive HT424, developes higher fatigue properties. For a double lap joint for graphite-steel only data on Shell 951 adhesive are available. This adhesive has properties similar to AF126-2. Test variables were also t_1/t_2, where t_1 = thickness of outer adherends (metal) t_2 = thickness of the graphite composite and t_a = adhesive thickness. The best shear values were achieved with $t_1 = .50\ t_2$ and $t_a = .10\ t_2$. Particularly significant was the effect of t_a/t_2 with $t_a = .01\ t2$ having the lowest values.

For the joint in this analysis, the following dimensions apply,

Average adherend thickness

$t_1 = .30$

$t_2 = .45$

Adhesive thickness (carrier system)

$t_a = .010$ in.

Overlap length

$L_a = 4$ in.

Then

$$L_a/t_2 = 8.9, \quad t_1/t_2 = .67, \quad t_a/t_2 = .022$$

From the design curve for the graphite/titanium joint in the "Design Guide" follows that the allowable static shear stress for the Shell 951 adhesive for these parameters is approximately 2,600 psi.

With the results of boron-steel cycling tests as reference, the allowable static shear stress for a HT424 type adhesive with a graphite adherend is:

$$F^{su} = 2,432 \cdot 2,600/3,990 = 1,585 \text{ psi}$$

and the allowable R = - 1.0 stress for $2.5 \cdot 10^6$ cycles,

$$F^{s} = 1,585 \cdot .27 = 428 \text{ psi}$$

4. <u>Estimate of Applied Stress</u> - The most critical stresses in the bonded joint are the bending shear stresses. Due to the rigidity of the joint design in resisting compressive loads, the stress analysis cannot be

based on the assumption, that the neutral axis is the longitudinal axis of the hub. Rather it must be assumed, that the neutral axis for adhesive shear analysis involving bending moment is between the central axis and the bonded joint, probably close to the bonded joint.

For this study the assumption is made that the effective neutral axis is at 75% of the mean radius of the bonded joint. This concept is shown in Figure 2-16. With reference to the neutral axis the section moment of inertia is:

$$I_1 = .049\ (13.93^4 - 12.65^4) + .7854\ (13.93^2 - 12.65^2) \cdot 5.22^2$$

$$= 590.26 + 728.11$$

$$= 1318.37 \text{ in.}^4$$

Then the minimum section modulus is:

$$Z_{min} = \frac{1318.37}{6.965 + 5.22} = 108.2 \text{ in.}^3$$

The hypothetical maximum load per 1 in. width to be resisted by adhesive in the joint then becomes:

$$P_1 = \frac{79,274 \cdot 7.03}{108.2} \cdot .64$$

$$= 3,296 \text{ lbs/in}$$

Consequently the average longitudinal shear stress due to bending in the double lap joint is:

$$f^s_{av(B)} = \frac{3,296}{1 \cdot 8} = 412 \text{ psi}$$

The circumferential shear stress due to torque is:

$$f^s_{av(T)} = \frac{480,000 \cdot 2}{13.2^2 \ \pi \cdot 8} = 219 \text{ psi}$$

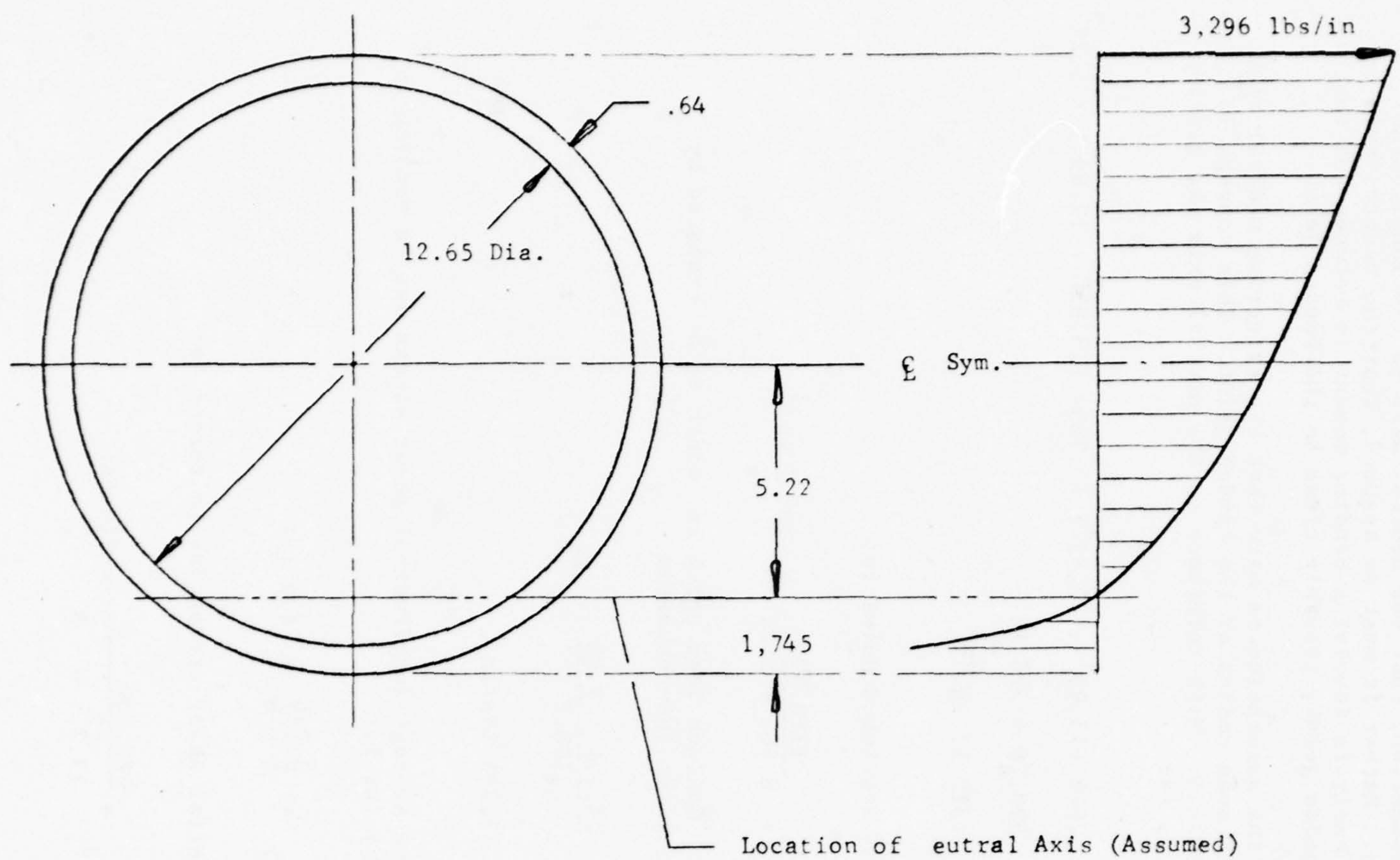


Figure 2-16 Load Distribution in Bonded Joint

The resultant shear stress is determined from the resultant shear load acting on the unit shear area:

$$f^{s}_{RES} = 1/8 \sqrt{(412 \cdot 8)^2 + (219 \cdot 8)^2}$$

$$= 466 \text{ psi}$$

This value is somewhat larger than the estimated allowable shear stress of 428 psi, but for this approximate analysis can be considered acceptable.

5. Analysis of the Threaded Joint - The threaded joint between the two concentric flanges must transmit 50% of the longitudinal bending and torque load. The longitudinal load is transmitted by interlocking of the threads and the torque load by shear of the adhesive which is between the threads.

The dimensions of the threads are:

Pitch = .125 in.

Pitch Diameter = 13.37 in.

Thread Outer Diameter = 13.50 in.

The longitudinal load tends to shear the thread at the pitch diameter.

The area of one inch arc length of one thread is:

$$A_1 = .125/2 = .065 \text{ in}^2/\text{in}$$

At a total thread length of .75 in. there are six threads with a total area of:

$$A_2 = .065 \cdot 6 = .375 \text{ in.}^2$$

The area of all threads (60° thread angle) in contact is:

$$A_3 = \pi \cdot 13.37 \cdot .75 \cdot 2 = 63 \text{ in.}^2$$

Stresses in the thread are:

Shear due to the longitudinal loads and possible simultaneous side load on the guide flange is:

$$f^{s}_{(b)} = \frac{3296/2 + 3840}{.375} = 14,635 \text{ psi}$$

The allowable ultimate shear stress in steel is $F^{su} = 56,000$ psi. Therefore the margin of safety is high.

The torque shear in the adhesive between the threads is:

$$f^{s}_{(T)} = \frac{480,000 \cdot 2}{2 \cdot 13.37 \cdot 63} = 570 \text{ psi}$$

The allowable ultimate shear stress in the adhesive was estimated earlier to $F^{SU} = 1,585$ psi. Therefore the margin of safety is:

$$\text{M.S.} = \frac{1,585}{570} - 1 = +1.78$$

Fatigue cycle allowable stress is considerably higher than the 428 psi reported earlier because the number of cycles for full reversal of the torsional load is significantly less than $2.5 \cdot 10^{6}$ and the above applied shear stress of 570 psi will have an adequately safety factor against fatigue failure.

6. Weight Analysis of"Concept B"- The calculated weight for the composite hub design "Concept B" shown in Figure 2-15 is 213.6 lbs. Of that weight only 19.6 or 9% is fibrous composite materials and 194.0 lbs. or 91% is steel.

The weight of the present steel hub is 303 lbs. Weight savings is then 303 - 213.6 = 89.4 or 29%.

7. Comparison of Hub Design "Concept A" and "Concept B" - The large difference in weight is the most significant factor in the comparison of the two design concepts. The all composite design weights 113.25 lbs. and the steel composite hybrid design 213.62 lbs. The main advantage of "Concept B" is the relative simplicity in manufacturing of the composite cone section. However the steel casting, including the machining operation is very complex. Structurally, both concepts are equivalent,

however the dependence on bonding in "Concept B" is undesirable. "Concept A" depends more on fiber strength, than on adhesive strength, which is the desirable method for designs using composite materials.

The design in "Concept A" is decidely more attractive than "Concept B".

2.3.2.6 Sprocket

The drive torque is transmitted from the sprocket tooth to the track connector link by contact pressure. Foreign particles, like sand and rocks as well as water and oil might be present between the contact surfaces. As estimated earlier, the sprocket tooth must also resist bending and shear in the direction of the torque load and some unspecified side load.

As determined previously, the maximum load per tooth consists of track pretension and track drive load and is 25,910 lbs. normal to the contact surface.

The radius of the tooth in the contact area is 0.97 in. and of the connector .94 in., leaving a theoretical gap of .03 in. The length of the contact is 1.88 in.

The contact load causes the surface to deform, resulting in a contact area with the length "L" (here 1.88 in.) and the width "b". An approximate value of "b" can be estimated using the formulas developed by Hertz, assuming Poisson's ratio to be 0.3. For two cylinders of the same material the deformed width is (convex in concave):

$$b = 2.185 \sqrt{\frac{2P}{LE} \Big/ \left(\frac{1}{R_2} - \frac{1}{R_1}\right)}$$

where

R_1 = radius of the large cylindrical surface (tooth)

R_2 = radius of the small cylindrical surface (connector)

$$P_3 = 15{,}000 \cdot .65 \cdot .45 = 4{,}388 \text{ lbs.}$$

For this application the theoretical contact width is:

$$b = 2.185 \sqrt{\frac{2 \cdot 25{,}910}{1.88 \cdot 29 \cdot 10^6} \Big/ \left(\frac{1}{.94} - \frac{1}{.97}\right)}$$

$$b = .371 \text{ in.}$$

The average surface pressure is:

$$p = \frac{25{,}910}{1.88 \cdot .371} = 37{,}148 \text{ psi}$$

The maximum surface pressure is:

$$p_{max} = .583 \sqrt{\frac{P}{L} \cdot \left(\frac{1}{R_2} - \frac{1}{R_1}\right) \Big/ \frac{2}{E}}$$

$$p_{max} = .583 \sqrt{\frac{25{,}910}{1.88} \cdot \left(\frac{1}{.94} - \frac{1}{.97}\right) \Big/ \frac{2}{29 \cdot 10^6}}$$

$$p_{max} = 47{,}272 \text{ psi}$$

For safe operation, this value should be less than the proportional limit compressive stress of the material.

The present steel material is either a casting class 80-50 through 105-85, or a forging 4150, with a surface hardness Rockwell C55 to C60 for .25 in. depth and C50 for a maximum of .62 in. depth. The material properties for the connector is equivalent to steel forgings, such as 4140, hardened to Rockwell C 40/45 hardness. Consequently the material in the connector is softer than the tooth material. The proportional limit for this type steel is approximately 150,000 psi, considerably higher than the applied maximum surface pressure.

Of further importance is the shear stress, which reaches its maximum just below the contact surface. This value is 30% of the compressive maximum, or, 12,703 psi.

For this study it was important to find the contact pressures for cylinders of different materials. For this purpose the modulus of elasticity of the material in the direction of the pressure must be known.

The contact width of the nested cylinders is:

$$b = 2.185 \sqrt{\frac{P}{L} \quad \left(\frac{1}{E_1} + \frac{1}{E_2} \right) \Big/ \left(\frac{1}{R_2} - \frac{1}{R_1} \right)}$$

and the maximum compressive stress:

$$p_{max} = .583 \sqrt{\frac{P}{L} \quad \left(\frac{1}{R_2} - \frac{1}{R_1} \right) \Big/ \left(\frac{1}{E_1} + \frac{1}{E_2} \right)}$$

Consideration of Poisson's ratio values different from 0.3, may be neglected, because the effective compressive width depends on the magnitude,

$$(1 - \nu^2)$$

and would not vary more than ±5% for fiberous composite materials.

For this analysis it was assumed that the sprocket is made of Acetal molding compound while the connector link is steel.

Acetal molding compound designated Celcon GC-25A (25% glass filled) by Celanese Plastics Company has high mechanical properties and good thermal and chemical resistance. One of the applications for this material is automotive gears.

The reported properties are:

E^t, E^c = $1.2 \cdot 10^6$ psi

E^b = $1.1 \cdot 10^6$ psi

F^{tu} = approximately 20,000 psi (estimated)

F^{ty} = 16,000 psi

$F^{prop.}$ = approximately 13,000 psi (estimated)

F^{cu} = approximately 30,000 psi (estimated)

F^b = 23,500 psi

F^s = 13,000 psi

γ = .0574 lb/in^3

The contact width is then:

$$b = 2.185 \sqrt{\frac{25,910}{1.88} \left(\frac{1}{29 \cdot 10^6} + \frac{1}{1.2 \cdot 10^6}\right) \Big/ \left(\frac{1}{.94} - \frac{1}{.97}\right)}$$

$b = 1.32$ in.

And the maximum contact pressure:

$$P_{max} = .583 \sqrt{\frac{25,910}{1.88} \left(\frac{1}{.94} - \frac{1}{.97}\right) \Big/ \left(\frac{1}{29 \cdot 10^6} + \frac{1}{1.2 \cdot 10^6}\right)}$$

= 13,309 psi

Maximum shear stress = $13,309 \cdot .30 = 3,993$ psi

The contact width and the maximum pressure all affected by the difference between radii R_1 and R_2. When R_1 and R_2 approach the same value "b" increases and p_{max} decreases. The maximum possible contact arc between tooth and connector is limited by the geometry of the tooth and is equal to 67° (Figure 2-11) or 1.10 in., which is less than the required contact width of 1.32. This is probably acceptable, however the contact pressure is greater than the estimated proportional limit of the tooth material and in order to make a realistic decision on feasibility of this design concept, properties for Celcon under repeated contact pressure must be developed.

1. Check of Bending and Shear Stresses in Tooth - As shown in Figure 2-11 the contact load is at an angle relative to symmetry line of the tooth and is pointing to the base of the tooth resulting in a small bending moment. Therefore for the analysis of the tooth in bending, the tangential component of the drive force was selected as the most critical possible design load.

It is assumed that the drive force of 9,794 lbs/tooth is acting at the pitch-radius and subjects the tooth to shear and bending.

Shear stress in the cross section at the pitch radius,

Width = 1.88 in.

Height = 2.50 in.

$$f^s_{max} = \frac{3 \cdot 9,794}{2 \cdot 1.88 \cdot 2.50} = 3,126 \text{ psi}$$

Bending stress at root radius,

Width = 1.88 in.

Height = 4.25 in.

Moment arm = 1.93 in.

$$f^b_{max} = \frac{9,794 \cdot 1.93 \cdot 6}{1.88 \cdot 4.25^2} = 3,340 \text{ psi}$$

These stress levels are acceptable for Celcon.

From this analysis it is concluded that Celcon is a feasible material for the sprocket. However, an unknown area is wear under operational conditions. This must be determined through tests.

The weight of the sprocket, molded from Celcon is 110 · .0574/.283 = 22.3 lbs. or 20% of the weight of the present steel sprocket.

Another composite material suitable for the sprocket is a laminate of glass or graphite fabric, either as a wet molded part or machined from a flat laminate.

The in-plane properties of graphite laminates are:

		Warp Direction	Isotropic: 0°/60°/120°
E^t, E^c	psi	$10 \cdot 10^6/8 \cdot 10^6$	$6 \cdot 10^6$
F^{tu}	psi	93,000	70,000
$F^{prop.}$	psi	55,000 (estimated)	40,000 (estimated)
F^c	psi	88,000	70,000
F^s	psi	19,000	19,000 (estimated)
ply	psi	.008	.008
γ	lb/in^3	.057	.057
$F^{prop.}_{cycl.}$	(60% static) psi	-	24,000
$F^s_{cycl.}$	(50% static) psi	-	9,500

$2.5 \cdot 10^6$ cycles

With the above isotropic properties and the previous tooth and connector dimensions the following stresses were calculated:

Contact Width:

$$b = 2.185 \sqrt{\frac{25,190}{1.88} \left(\frac{1}{29 \cdot 10^6} + \frac{1}{6 \cdot 10^6} \right) \Big/ \left(\frac{1}{.94} - \frac{1}{.97} \right)}$$

$$b = .634 \text{ in.}$$

or 58% of available arc length.

Contact Pressure:

$$p_{max} = .583 \sqrt{\frac{25,190}{1.88} \left(\frac{1}{.94} - \frac{1}{.97} \right) \Big/ \left(\frac{1}{29 \cdot 10^6} + \frac{1}{6 \cdot 10^6} \right)}$$

$$p_{max} = 27,759 \text{ psi}$$

Shear below the contact surface:

$$f^s = 27,759 \cdot .30 = 8,328 \text{ psi}$$

Since the bending and the shear bending stresses are the same as calculated previously for the Celcon design it is concluded that all stresses are within the design allowables and that a sprocket made from a graphite fabric/epoxy laminate is acceptable from a strength standpoint. However, an unknown area is wear under operational condition, same as for the Celcon material. In addition delamination can occur in a laminate subjected to edgewise loading and may require special attention of the design of the tooth. The weight of a graphite composite sprocket is similar to the sprocket made from Celcon.

Because of possible delamination problems with the graphite laminate concept it is felt that the molded Celcon concept is better suitable for this application. It is also possible that a Celcon compound with higher glass fiber content could be obtained (higher than 25%) which would improve the properties of this material.

2.3.3 Manufacturing Analysis

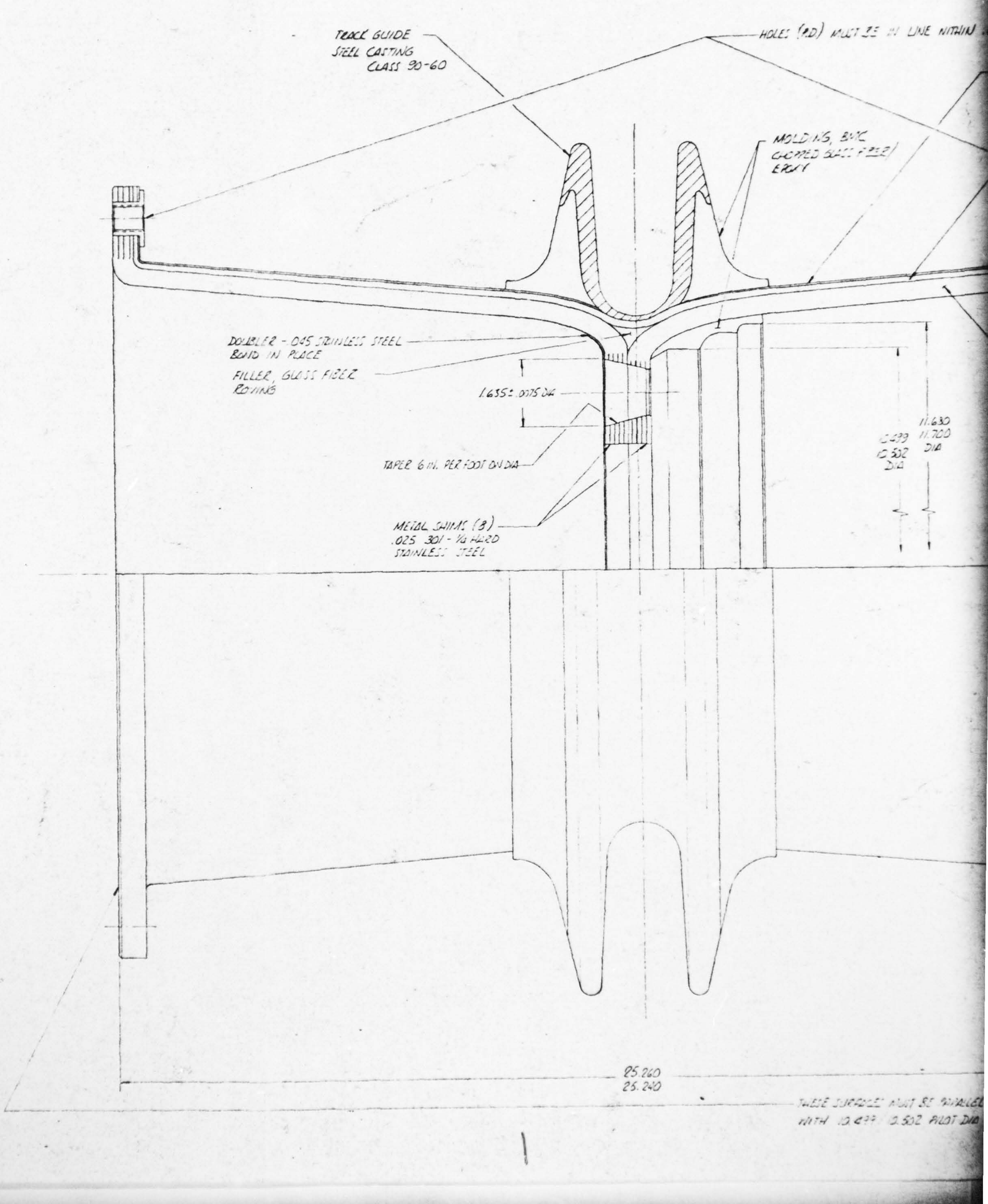
Drawings of the drive wheel hub and sprocket are shown on Figures 2-17 and 2-18. A suggested method for the manufacture of the hub is shown on Figure 2-17. The U-shaped ring (track guide) is a steel casting. It may be cast as one piece or two halves welded together. The track guide is supported by gusset rings molded from short fiber bulk molding compounds. The rings must be molded in place. A concept for the molding of the rings is shown on Figure 2-17. The female mold is segmented and shaped to fit the outside contour of the track guide. The ram is shaped to mold the finished outer contour of the gusset ring. The inside contour can be molded net or slightly undersized and machined to final shape. Net molding of the inner contour would require molding of one gusset ring at a time. If molded undersize, sufficient space must be provided between the inside contour of the track guide and the tool to allow the molding compound to flow into the opposite cavity in the mold.

The layup of the outer layers of glass fabric and the 0° graphite fibers is performed on a female mold. This is shown as Step 2 on Figure 2-17. The mold is segmented to allow inserting the track guide and the gusset rings. The layup is vacuum bagged and compacted and excess resin is bled out but the material is not cured at this time.

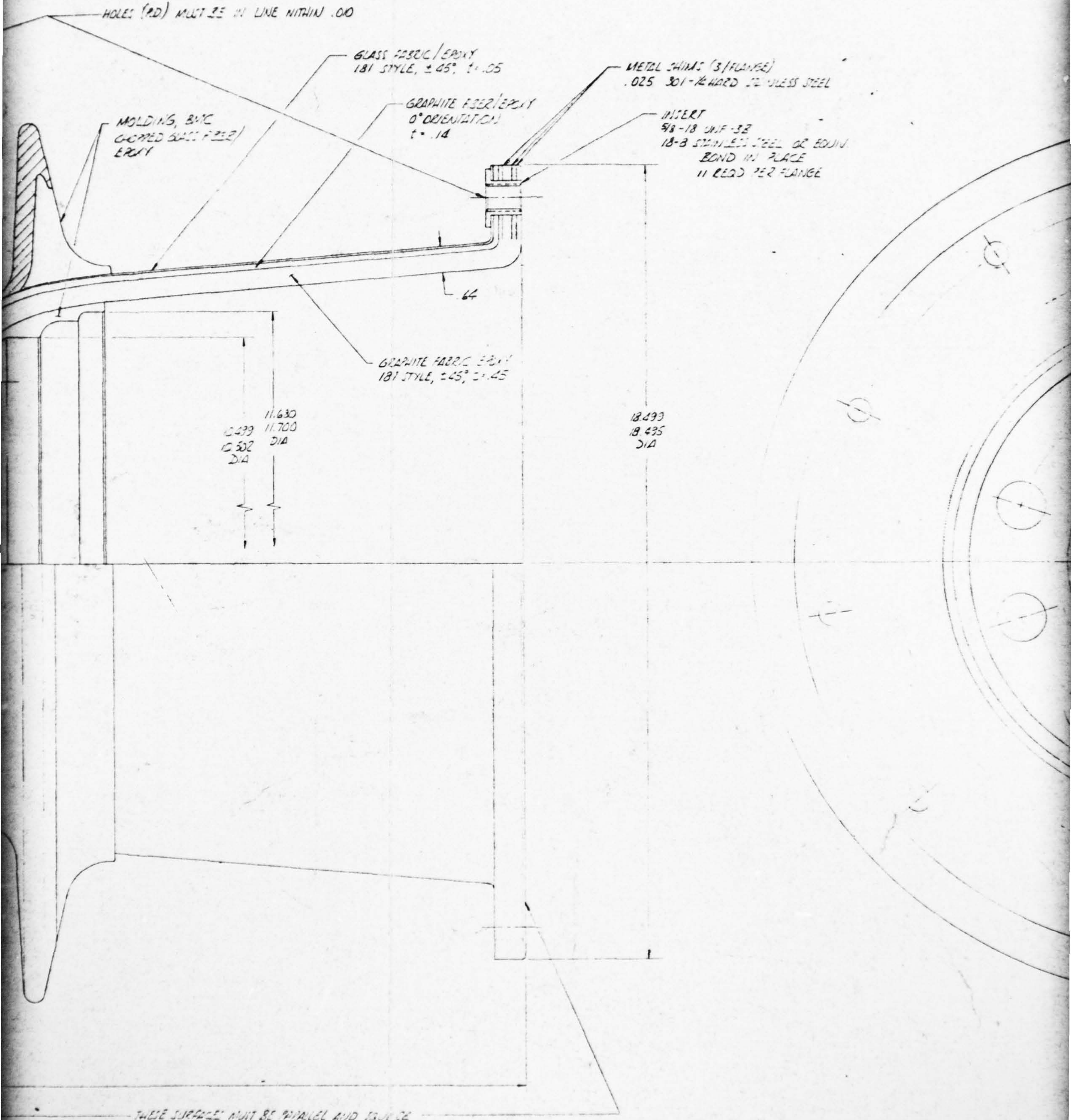
Step 3 on Figure 2-17 shows the layup of the ±45° graphite fabric material on male mandrels. There are two mandrels, one for each side of the hub. Because of the shape of the layup, the fabric must be cut in segments. The segments are butted and segments in adjacent layers are placed so that the joints are staggered. The layup is compacted and prebled but not cured. The mandrels are then placed in the female mold with the 0° graphite fiber layup and the assembly is cured. Curing pressure is provided by a rubber jacket on the mandrel. The thickness of the rubber jacket must be selected so that the correct pressure is applied during the thermal expansion of the rubber.

Close tolerance dimensions such as the outside diameter of the flanges, flange faces, internal configuration for seals and bolt hole locations must be machined in the cured part.

The sprocket design is shown on Figure 2-18. This design is for a sprocket laminated from graphite fabric but the configuration is the same for a sprocket molded from sheet molding compound. The sprocket is molded in a compression mold to net dimensions with machining limited to the mounting holes.



TRACK GUIDE
STEEL CASTING
CLASS 90-60
HOLES (P.D) MUST BE IN LINE WITHIN
MOLDING, BMC
CHOPPED GLASS FIBER/
EPOXY
DOUBLER - .045 STAINLESS STEEL
BOND IN PLACE
FILLER, GLASS FIBER
ROVING
1.635 ± .0075 DIA
TAPER 6 IN. PER FOOT ON DIA
METAL SHIMS (8)
.025 301 - ½ HARD
STAINLESS STEEL
11.630
11.700
DIA
10.499
10.502
DIA
25.260
25.240
THESE SURFACES MUST BE PARALLEL
WITH 10.499 10.502 PILOT DIA



HOLES (P.D) MUST BE IN LINE WITHIN .010
GLASS FABRIC / EPOXY
181 STYLE, ± 45°, t = .05
METAL SHIMS (3/FLANGE)
.025 301-½ HARD STAINLESS STEEL
GRAPHITE FIBER/EPOXY
0° ORIENTATION
t = .14
INSERT
5/8-18 UNF-3B
18-8 STAINLESS STEEL OR EQUIV.
BOND IN PLACE
11 REQD PER FLANGE
MOLDING, BMC
CHOPPED GLASS FIBER/
EPOXY
.64
GRAPHITE FABRIC EPOXY
181 STYLE, ±45°, t = .45
11.630
11.700
DIA
10.499
10.502
DIA
18.499
18.495
DIA
THESE SURFACES MUST BE PARALLEL AND SQUARE
WITH 10.499-10.502 PILOT DIA WITHIN .010 T.I.R.

2

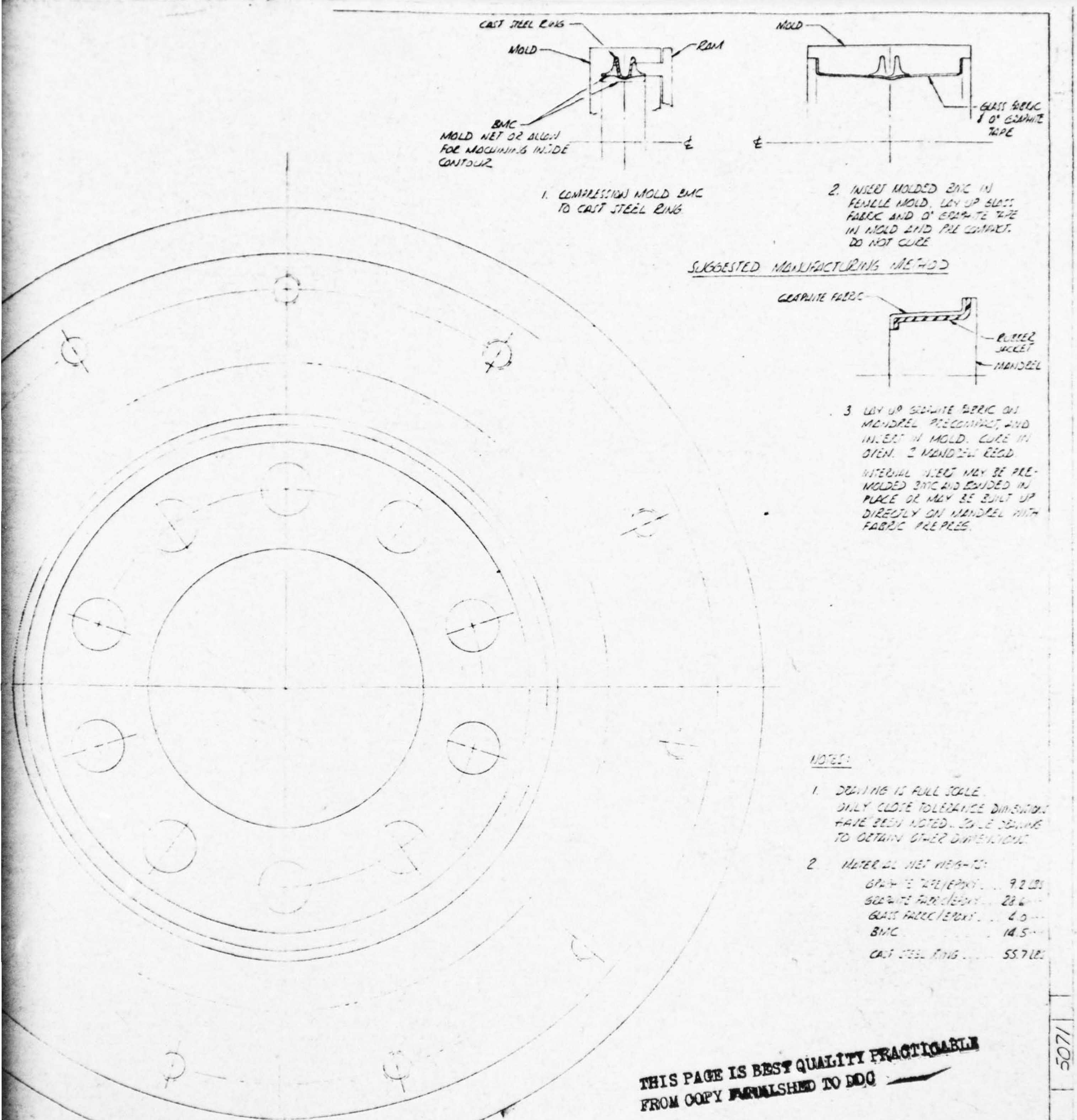
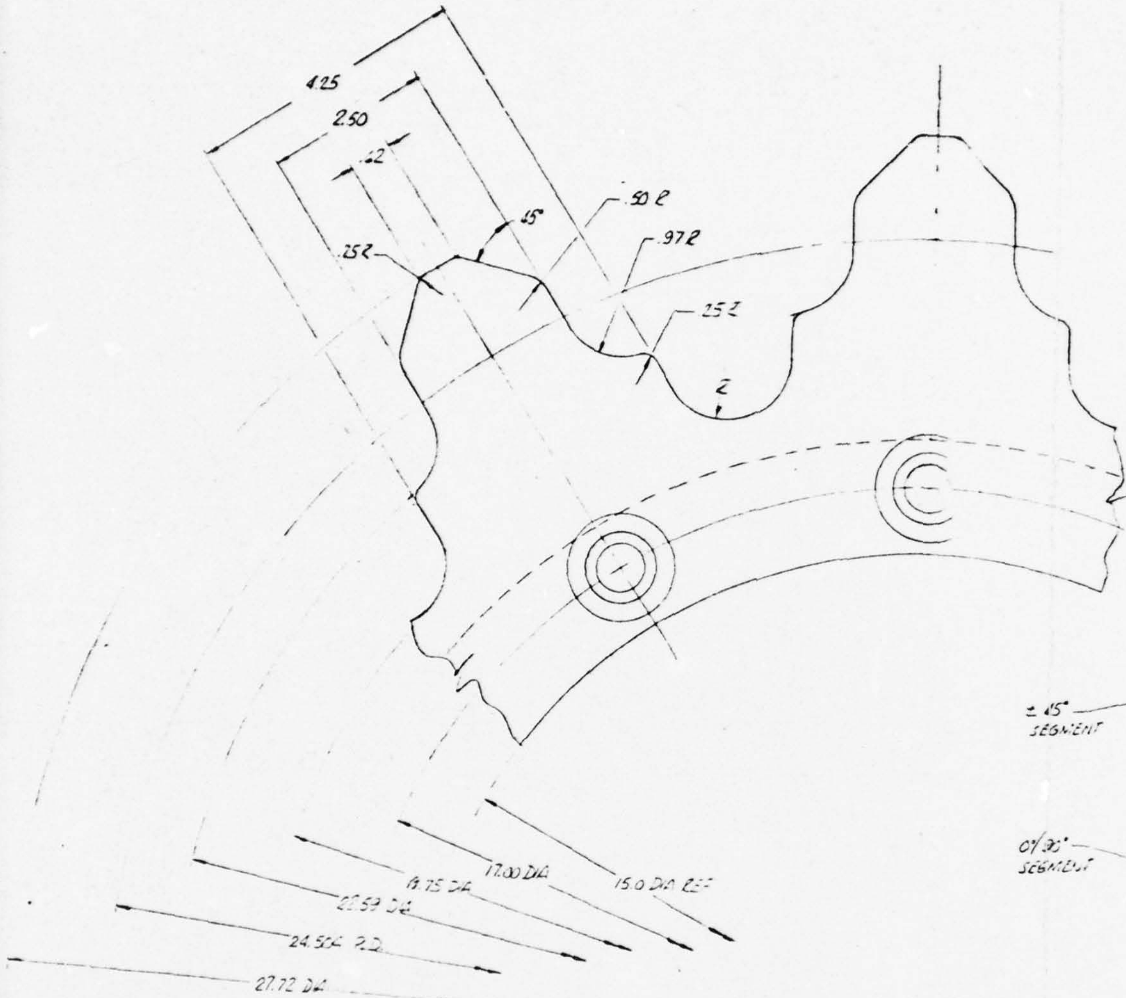


Figure 2-17

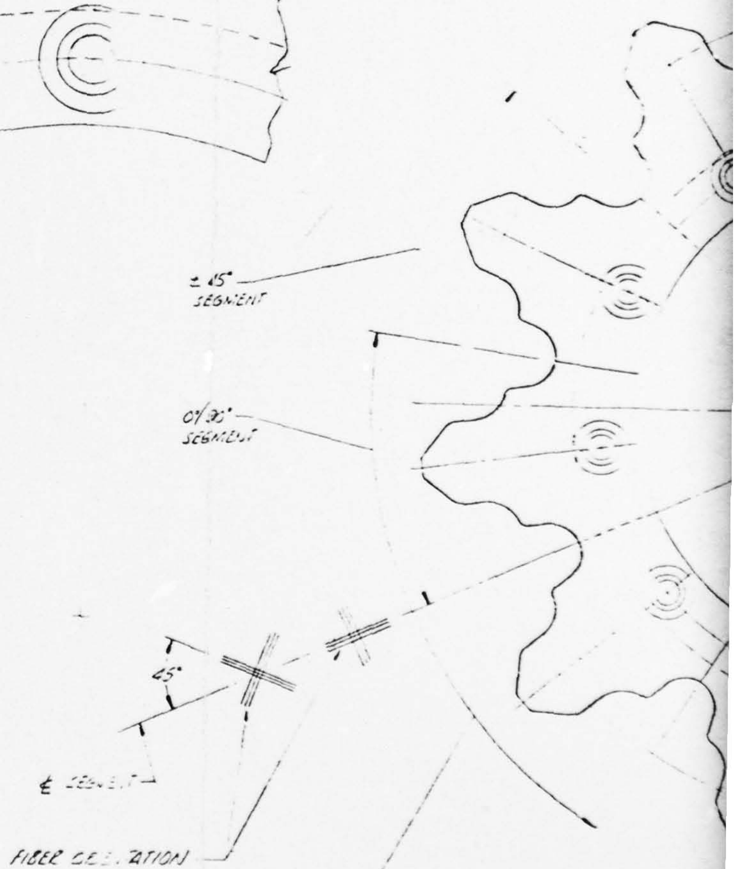
Hub, Composite Material

PART NO.	PART NAME	
DRAFT [illegible]		Riggs Engineering Corp. — San Diego, California
CHECK		
DESIGN		HUB, FINAL DRIVE - COMPOSITE MATERIAL (AMMRC)
STRESS		
PROCESS		
TOOLING		
APPROVED		
SIZE 2	CODE IDENT NO	DRAWING NO 5071
SCALE 1/1	CALC WT.	SHEET 1 OF 1

3



TOOTH DETAIL
SCALE 1:1



CUT FABRIC PREPREG IN SEGMENTS COVERING
TWO TEETH AS SHOWN. FIBER ORIENTATION TO
BE 50% 0°/90° & 50% ±45° WITH RESPECT TO
℄ OF SEGMENT. PLIES TO BE FULLY DISPERSE
STAGGER ADJACENT PLIES ONE TOOTH PITC
AS SHOWN.

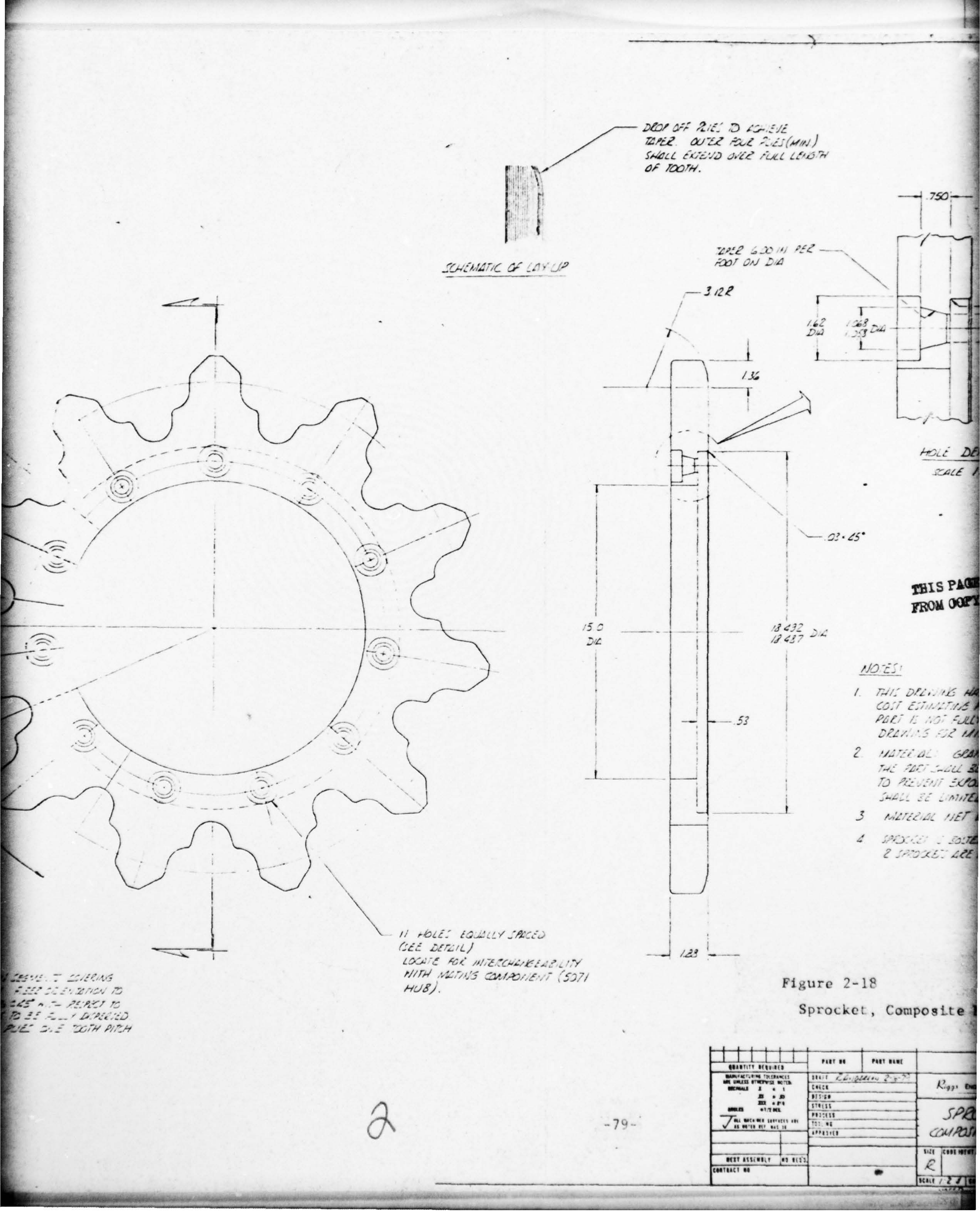


Figure 2-18

Sprocket, Composite

2

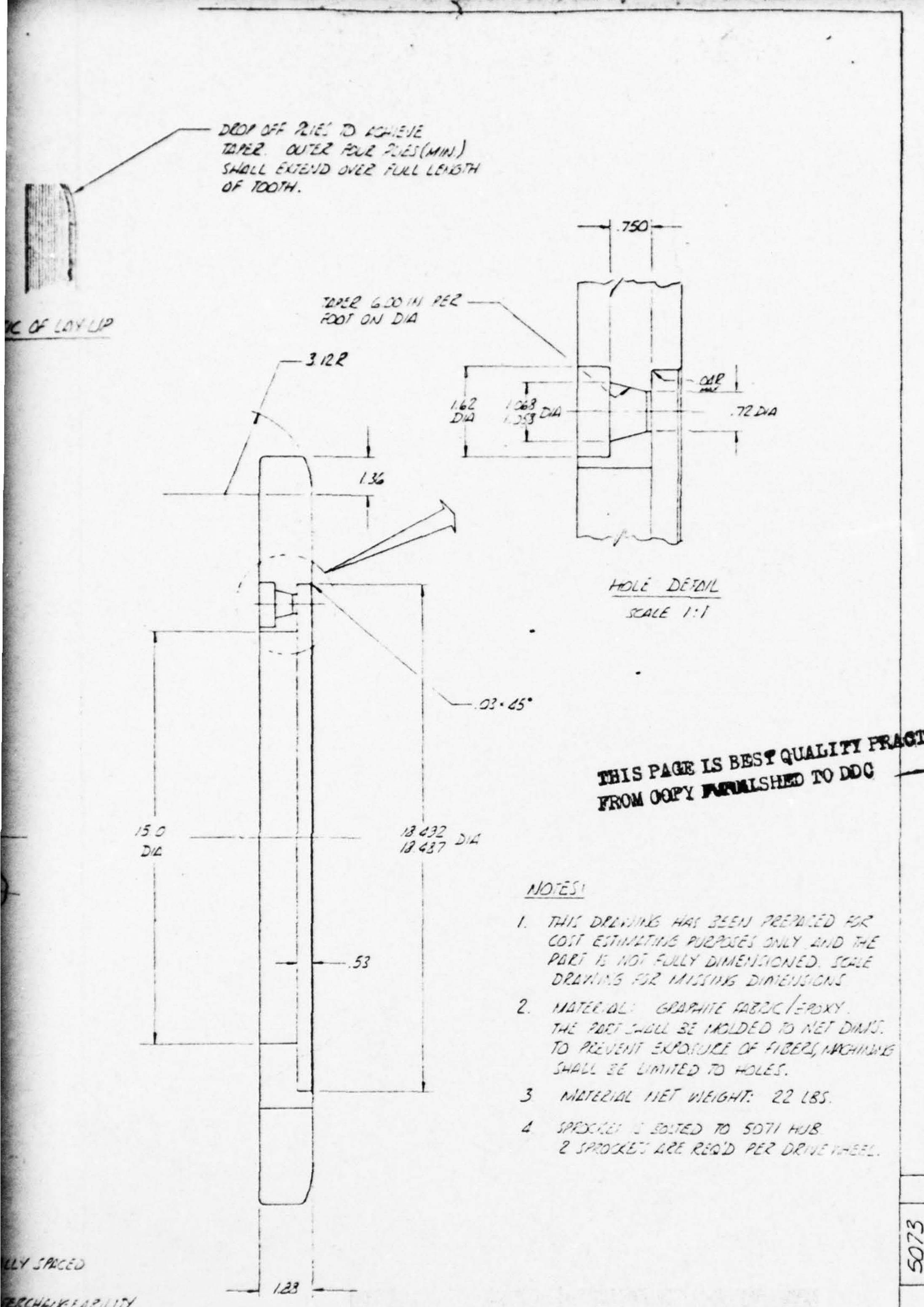


Figure 2-18

Sprocket, Composite Material

QUANTITY REQUIRED | PART NO | PART NAME

Riggs Engineering Corp.

SPROCKET - COMPOSITE MAT'L (AMMRC)

SIZE R | DRAWING NO 5073 | SHEET 1 OF 1

3

2.3.4 Manufacturing Costs

The design of the drive wheel hub and the sprocket shown in Figures 2-17 and 2-18 are suitable for production in quantities less than 500 units. For quantities of several thousand units further studies in fabrication techniques must be conducted and new low cost methods must be developed in order to produce a cost effective part. To date application of advanced fibrous composite materials to structural components of complex shapes have been mainly in the aerospace industry where production quantities are relatively small. Large quantities of composite parts used in the automotive industry are manufactured from various molding compounds which do not contain oriented continuous fibers and are not critical, load carrying structures.

Estimated costs to manufacture the drive wheel hub and sprocket are shown in Table 2-6 for quantities of 10, 100 and 10,000 units. Costs for the sprocket are based on laminated graphite fabric/epoxy. Molded from Celcon or some other molding compound the cost of the sprocket would be reduced by approximately 50%. To determine the suitability of these materials, more material test data should be obtained, especially on wear of composite material gears presently in use in industry.

Table 2-6 Drive Wheel Manufacturing Costs

Hub, Drive Wheel (Figure 2-17)

Development Engineering and Manufacturing Drawings $4500.

Quantity	10	100	10,000*
	Cost/Unit, $		
Material	4870	2160	1599
Fabrication and Quality Control	2064	1698	973
Prod. Engineering and Program Adm.	978	139	9
Tooling	1610	239	17
Total:	9552	4236	2598

Sprocket (Figure 2-18)

Development Engineering and Manufacturing Drawings $1000.

Quantity	10	100	10,000*
	Cost/Unit, $		
Material	1316	1108	779
Fabrication and Quality Control	522	409	224
Prod. Engineering and Program Adm.	694	83	6
Tooling	897	138	6
Total:	3429	1738	1015

*Fabrication Rate: 1000 units/year

2.4 Track Support Roller

2.4.1 Baseline Requirements

2.4.1.1 Description

The support rollers support the upper (return) portion of the track loop between the drive wheel and idler wheel. There are three identical roller assemblies on each side of the tank. Each roller assembly consists of two identical wheels mounted back to back on a hub with sufficient space between the wheels to allow clearance for the fork like track guide links mounted at the center of track between each track shoe. Each wheel has a solid rubber tire vulcanized to the rim.

2.4.1.2 Dimensional Requirements

The dimensional requirements for the support roller are given by Drawing No. 8706067, Wheel Assembly, welded construction. The diameter of the metal wheel is 12 in. and the overall diameter, including the solid rubber tire, type II spec. MIL-T-3100, is 13.56 in. The overall width of the wheel is 4.172 in., including the wear ring. The wheels are mounted back to back on a hub, Part No. 8762545 by means of six hexagon head cup screws.

2.4.1.3 Design Load Condition

The only design load requirement provided in Appendix D, Table I, Item (3) of the Specifications and Requirements is a radial compressive load of 2,500 lbs. caused by the weight of the track. A factor of two was added to this load to account for dynamic loads. No side loads are given. To estimate the endurance requirements the RPM of the support roller was calculated for a maximum vehicle speed of 30 mph.

$$N = \frac{30 \cdot 63,360}{\pi \cdot D \cdot 60} = 744 \text{ RPM}$$

The frequency of the radial load change from zero to maximum is then 12.4 per sec. The rotation also imposes centrifugal forces on the wheel structure.

Assuming the minimum test mileage requirement per MIL-T-3100: Item 4.5.58 of 3,000 miles, the minimum life expectancy is 3,000/30 = 100 hrs. or not less than $744 \cdot 60 \cdot 100 = 4.5 \cdot 10^6$ cycles.

The physical properties of the tire material per MIL-T-3100 are:

Shore A hardness	70 ± 10 durometer
Minimum tensile strength	1,900 psi
Minimum elongation	200%
Specific gravity	1.2 g/cm^3

2.4.1.4 Weight of the Wheel

The weight of the present wheel is given as 22 lbs. including rubber tire. There are six roller pairs per vehicle. The total weight per vehicle is then $12 \cdot 22 = 264$ lbs.

2.4.2 Feasibility Study

2.4.2.1 Introduction

The geometry of the roller makes it a suitable component for manufacture by compression molding using a short fiber molding compound. This is a relatively inexpensive manufacturing method. The wheel including disc and rim can be molded in a single operation to its final configurations without any machining required after the molding operation. Even the mounting holes can be included in the molding.

Other manufacturing methods such as filamentary winding or layup of continuous fiber prepreg would produce a lighter wheel but it would also be costlier.

The following study analyzes two different molding compounds; stresses and deflections are compared with the present metal roller design.

2.4.2.2 Material Selection

Properties for two bulk molding materials are listed in Table 2-7 Both compounds consist of chopped fibers in an epoxy resin; the E-260H compound is glass fiber filled and E-21718 is graphite fiber filled. Both compounds are manufactured by Fiberite Corporation. The materials are used for compression molding in matched metal molds with a molding pressure of approximately 1,000 psi.

A strength reduction after $4.5 \cdot 10^6$ flexure cycles must be considered. It is estimated that for the glass filled compound the reduction may be

Table 2-7

Properties for Two Bulk Molding Compounds

	E260H (Glass)	E21718 (Graphite)
Molding Temp. °F	325	320
Bulk Factor	8-10	≤8
Density lb/in^3	.068	.051
Tensile, Ult. psi	27,000	20,000
Flex. Yield psi	55,000	40,000
Flex. Modulus 10^6 psi	4.1	5.5
Compression Ult. psi	42,000	28,000
Shear, Ult. psi (est.)	4,000	4,000
Izod., Notched ft. lbs/in	30	10
Water Abs. % 28 hrs. @50°C	.15	.80
Cost $/lbs.	low	high

as high as 70% and for graphite 60% of ultimate. Consequently the allowable flexure stress is .30 · 55,000 = 16,500 psi and .40 · 40,000 = 16,000 psi respectively. The allowable specific strength for the glass compound is 16,500/.068 = 242,650 in. and for the graphite compound 16,000/.051 = 313,725 in. Both materials meet the flexure strength requirements with the graphite compound possibly resulting in a lighter design. On the negative side for the E-21718 compound is its lower impact resistance, higher water absorption and higher cost.

It is reported that E-260H meets the impact strength per MIL-P-46069. This is not reported for the graphite compound, which is generally more brittle due to its higher modulus fiber than the glass fiber compound.

Fiberite E-260H glass fiber/epoxy molding compound was selected for the support roller.

2.4.2.3 Stress Analysis

For the analysis of the roller rim it is assumed that the rubber tire is compressed under load resulting in a parabolic pressure distribution over a contact width "b". The parabolic distribution is replaced by two concentrated loads through the centroids of one half of the parabolic area. The contact load is uniformly distributed over the width of the roller rim. It is resisted by the cylindrical rim and by the disc. For calculation of bending stresses in the rim, the rim is divided into narrow rings, each carrying a proportional amount of the total contact load. Each ring is supported in shear by the adjacent ring on the side facing the disc. The rim load is resisted by a shear flow of the magnitude,

$$S = \frac{Q}{\pi R_m} \cdot \sin \alpha \ (\mathrm{lb/in})$$

Where

Q = load on the rim ring

R_m = mean rim ring radius

α = angle between 0° and 180° beginning at the vertical load contact point

The parabolic contact pressure distribution is estimated by the Hertz formulas for contact between a cylinder and a flat plate. The contact width "b" is:

$$b = 2.154 \sqrt{\frac{Q \cdot R_o}{L} \left(\frac{1}{E_1} + \frac{1}{E_2}\right)}$$

Where

R_o = outer radius of tire = 6.78 in.

L = loaded tire width = 3 in.

E_1 = modulus of the track plate (steel)

E_2 = Modulus of the rubber tire. For a durometer of 70. E_2 = 2,100 psi was assumed.

Then

b = 3.53 in.

The distance from the centerline to the centroid of the half parabola .1875b = .66 in.

The maximum contact pressure is:

$$p_{max} = .592 \sqrt{\frac{Q}{L \cdot R_o} \Big/ \left(\frac{1}{E_1} + \frac{1}{E_2}\right)}$$

$$p_{max} = 300 \text{ psi}$$

The radial contact force and the circumferential shear causes bending stresses in the ring. The maximum bending moment is at the load contact point.

AD-A072 354 RIGGS ENGINEERING CORP SAN DIEGO CA F/G 11/4
FEASIBILITY OF COMPOSITE MATERIAL APPLIED TO TRACKED AND WHEELE--ETC(U)
JUN 79 B LEVENETZ, R N ANDERSON, K R BERG DAAG46-78-C-0066
UNCLASSIFIED AMMRC-TR-79-40 NL

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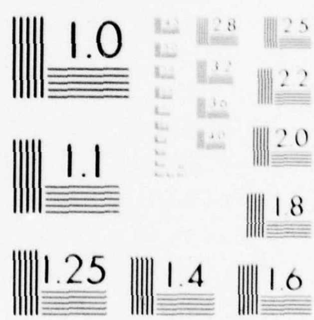


MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS 1963-A

One design concept of the support roller is shown in Figure 2- 19 . Th
contact pressure loads and the rim shear loads are plotted in Figure 2

The bending moments in the ring were calculated according to the formu
in Roark, Fourth Edition p. 178, Case 25. The same source was used to
determine circumferential tension, radial shear and radial vertical
change Δ R.

The bending moment between $\alpha = 0$ and $\alpha = \theta$ is:

$$M = QR \cdot K_1$$

and between $\alpha = \theta$ and $\alpha = \pi$

$$M = QR \cdot K_3$$

Where

$$K_1 = .23868 \cdot \cos \alpha - .50 \sin \theta + .15915 (K_2)$$

$$K_2 = \alpha \sin \alpha + \theta \sin \theta + \cos \theta - \cos \alpha \cos^2 \theta$$

$$K_3 = .23868 \cdot \cos \alpha - .50 \sin \alpha + .15915 (K_2)$$

Angle θ is 7° or .12217 rad

$$\sin \theta = .12187$$

$$\sin \theta = .99255$$

$$\cos^2 \theta = .98516$$

In the outer rim ring, $Q = 2,500/3.5 = 714$ lbs. and $R = R_{AV} = 5.8$ in.
Therefore $Q \cdot R = 4,141.2$ lb. in. The results are listed in Table 2-
and the bending moment distribution is plotted in Figure 2- 20.

The maximum moment is at $\alpha = 7°$ and is 775 in. lbs.

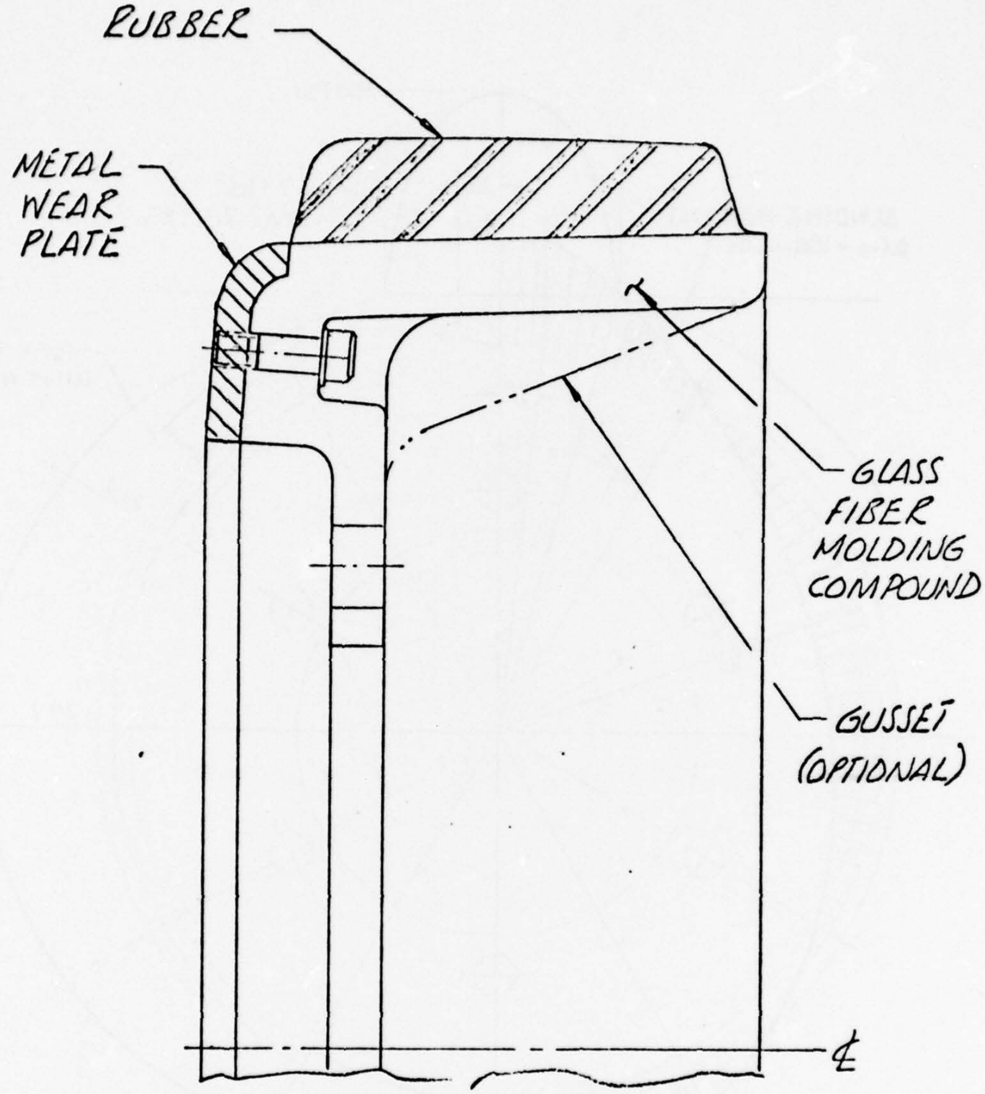


Figure 2-19 Support Roller Design Concept

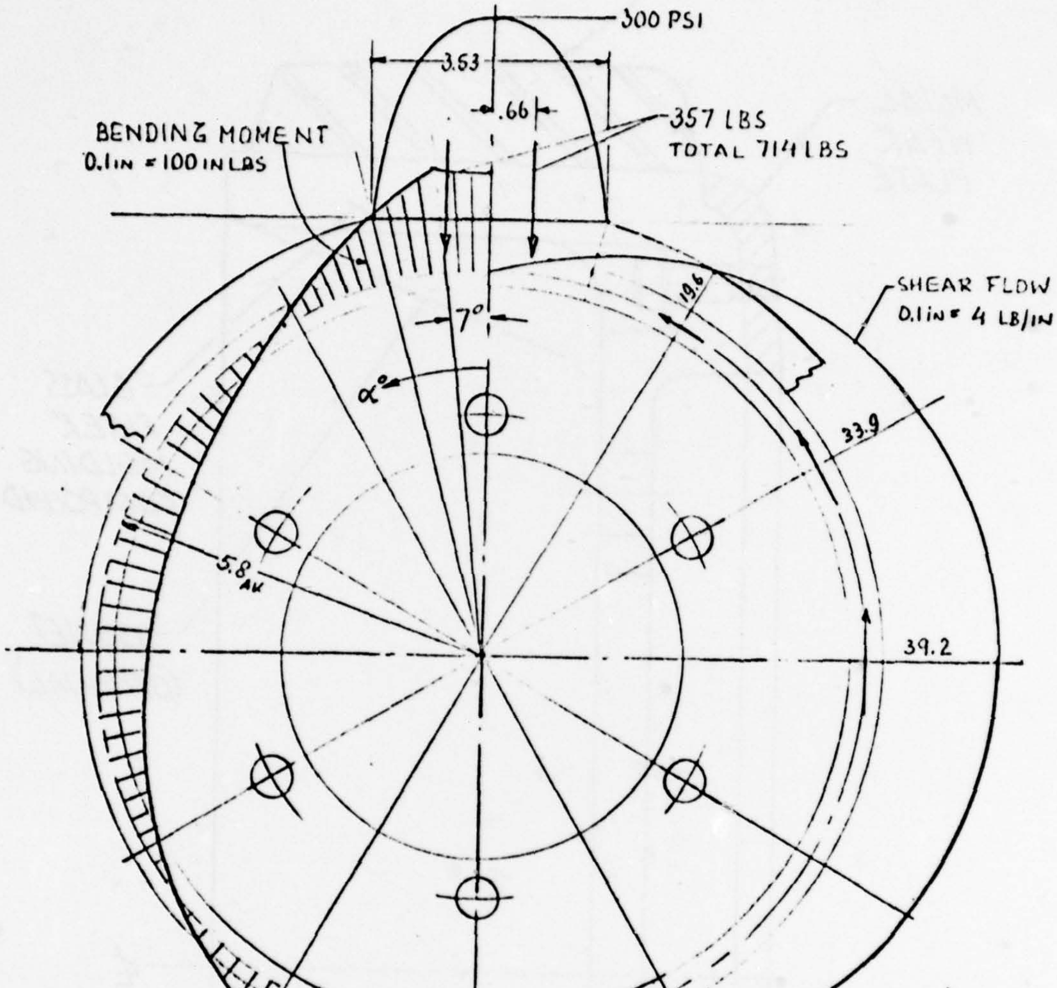


Figure 2-20 Load Distribution on Rim (1 in. Wide)

Table 2-8

Calculation of Bending Moments

$\alpha°$	α rad	Z $\sin \alpha$	U $\cos \alpha$	K_2	K_1	K_3	Mb in. lb.
0	0	0	1	.02228	.18128	-	751
5	.08727	.08716	.99619	.03723	.18276	-	757
7	.12217	.12187	.99255	.07073	.18722	.18722	775
15	.26180	.25582	.96593	.12360	-	.12081	500
30	.52360	.50000	.86603	.41606	-	.02292	95
60	1.04720	.86603	.50000	1.42177	-	-.08740	-362
90	1.57080	1.00000	0	2.57824	-	-.08967	-371
120	2.09440	.86603	-.50000	3.31383	-	-.02496	-103
150	2.61800	.50000	-.86603	3.16962	-	.04774	198
180	3.14159	0	-1	1.99260	-	.07844	325

With a rim wall thickness of .55 in., the bending stress in the rim is:

$$f_b = \frac{775 \cdot 6}{.55^2} = 15,372 \text{ psi}$$

The margin of safety for bending is then for $4.5 \cdot 10^6$ cycles:

$$\text{M.S.} = \frac{16,500}{15,372} - 1 = +.073$$

and for static material strength:

$$\text{M.S.} = \frac{55,000}{15,372} - 1 = +2.58$$

The circumferential tension in the rim due to the loading condition is calculated from:

$$T = Q \cdot K_4$$

Where

$$K_4 = .15915\ K_5 - .07958 \cos\alpha - .50 \sin\alpha$$

$$K_5 = \alpha \sin\alpha - .98516 \cos\alpha$$

The radial shear is calculated from:

$$S = Q \cdot K_6$$

Where

$$K_6 = .15915\ K_7 - .50 \cos\alpha$$

$$K_7 = \alpha\cos\alpha - .50 \sin\alpha + .98516 \sin\alpha$$

The calculated value for T and S are listed in Table 2-9.

Table 2-9

Calculation of Circumferential Tension and of Radial Shear

α°	K_5	$.15915\ K_5$	K_4	T_{lbs}	K_7	$.15916\ K_7$	K_6	S lbs.
0	-.98516	-.15679	-.23637	-159	0	0	-.5000	-357
7	-.9629	-.15325	-.29376	-210	.18038	.028708	-.4388	-313
15	-.8838	-.14066	-.34694	-248	.37845	.06023	-.4227	-302
30	-.4771	-.07593	-.39485	-282	.69618	.11079	-.32222	-230
60	+.4143	+.06594	-.406865	-291	1.12224	.178605	-.07139	-51
90	1.5708	.24999	-.25000	-179	.76209	.121286	+.121280	+87
120	2.3064	.36706	-.020165	-19	-.627036	-.09979	+.1502	+107
150	2.1622	.34411	+.16303	+116	-2.02468	-.32223	+.1108	+79
180	0.98516	.15679	+.23637	+169	-3.14159	-.49257	-0	-0

The maximum values result in the following stresses for a rim thickness of .55 in.

Tension $f_t = \frac{169}{.55} = 307$ psi

Compression $f_c = \frac{291}{.55} = 529$ psi

Shear $f_s = \frac{357}{.55} = 649$ psi

The maximum circumferential shear in the rim is at the section adjacent to the disc. The shear load is a portion of the contact load over the unsupported rim cylinder. This load is per Figure 2-20.

$$Q_1 = 2,500 \cdot 2.85/3.50 = 2,036 \text{ lbs.}$$

The maximum shear is at $\alpha = 90°$, $\sin \alpha = 1$

$$f_s = \frac{2,036}{\pi \cdot 5.7 \cdot .55} = 207 \text{ psi}$$

This analysis shows that the highest stress is expected in bending of the rim under contact pressure in its unsupported portion. All other stresses are low.

With the known modulus of elasticity, the radial deflection of the rim under contact load can now be calculated:

$$\Delta R = \frac{Q\ R^3}{E\ I} [.318 (.50 \cdot \cos\theta + .50\ \theta \cdot \sin\theta + .25 \cdot \sin^2\theta + .50) - .25 \cos\theta - .25\ \theta \sin\theta - .098]$$

For $\theta = 7°$, the bracketed value becomes = -.0293.

Then the radius change for $R_{av} = 5.8$ in.

$$\Delta R = -.095 \text{ in.}$$

The same analysis yields for the deflection of the present steel rim:

$$\Delta R_{STL} = -.112 \text{ in.}$$

This shows that the deflection of the molded composite rim is actually less than the steel rim.

The disc is loaded by the circumferential shear flow caused by the rim loads, the contact pressure acting over the flange and the mounting bolt forces resisting these loads

Maximum Shear = 207 psi

Contact Load 2,500 - 2,036 = 464 lbs.

Mounting Load = 2,500/6 = 417 lbs/bolt

The side loads transmitted by the track guide to the roller flange is not specified. It would tend to shear off a segment of the disc outside of the bolt circle diameter, probably through a length of approximately 3 in. For a disc thickness of .40 in. and an ultimate shear stress of 4,000 psi, the failing lateral load would be,

$$Q_L = 2 \cdot .40 \cdot 4{,}000 = 4{,}800 \text{ lbs.}$$

or almost twice the maximum vertical load on the support roller.

The mounting bolts are 5/8 in. diameter. The bolt bearing pressure is then:

$$p_{BR} = \frac{417}{.40 \cdot .63} = 1{,}655 \text{ psi}$$

By inspection the margin of safety for bearing stress is high.

2.4.2.4 Weight Estimate

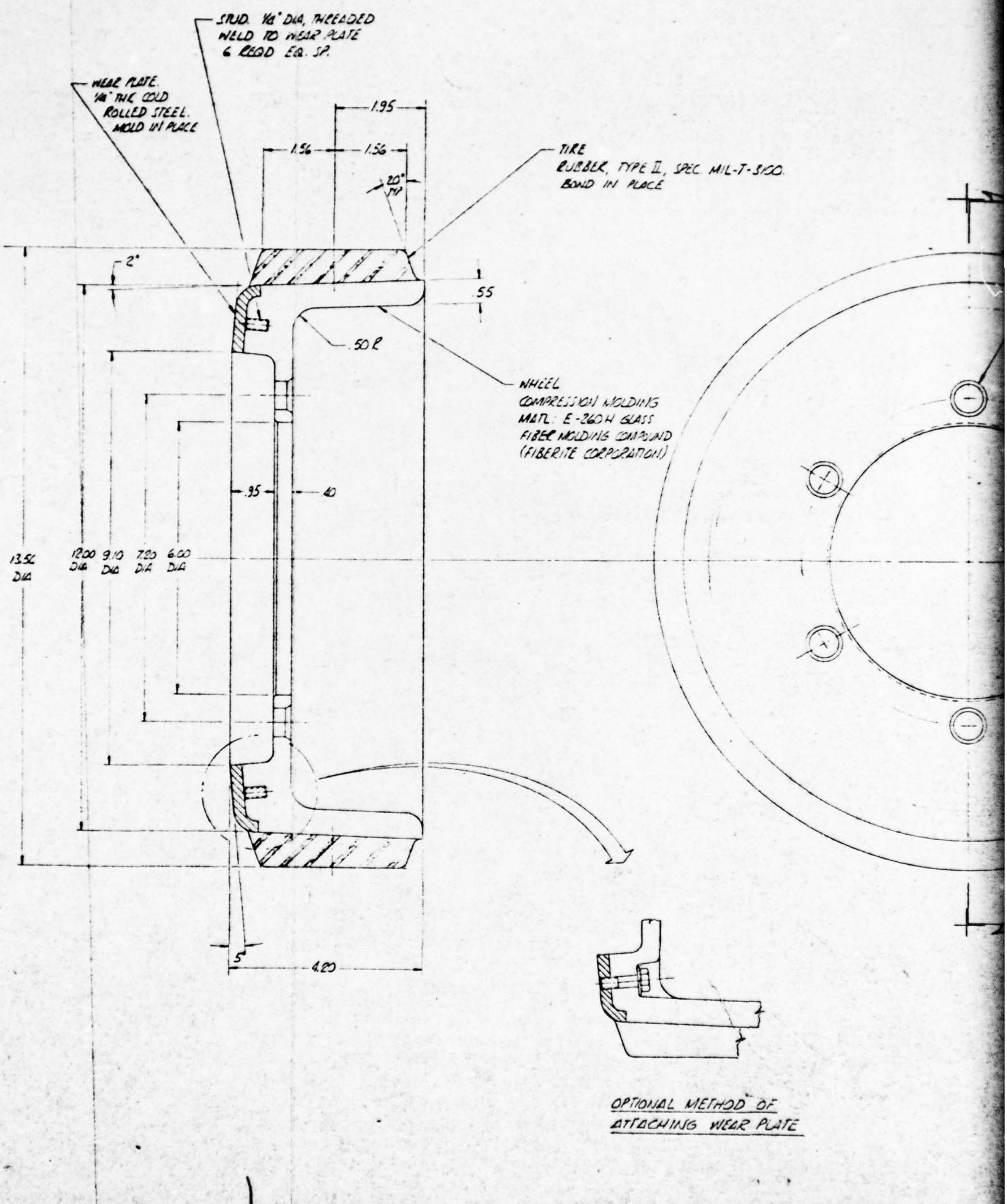
The weight of the design concept shown in Figure 2-19 was calculated to 15.7 lbs. This includes the wear ring which weights 3.8 lbs. and the rubber tire weighing 3.9. The present metal roller weighs 22 lbs. Weight savings is then 6.3 lbs. or 29%. The total weight saving for the vehicle would be $12 \cdot 6.3 = 76$ lbs.

2.4.3 Manufacturing Analysis

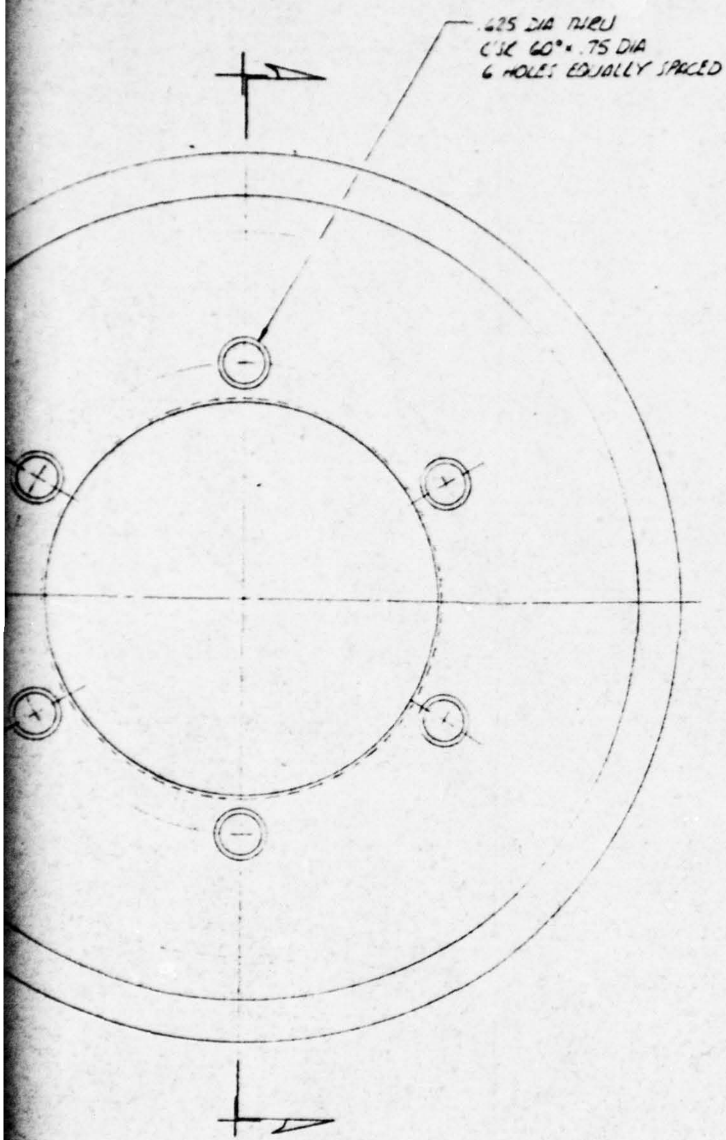
A drawing of the support roller is shown on Figure 2-21. The disc and rim structure is made from compression molded short glass fiber molding compound. The molding pressure is approximately 1,000 psi which requires matched steel dies. All features including mounting holes can be molded, no machining is required. The wear plate may be molded in place or bolted on as shown in the optional configuration. The rubber tire is manufactured separately and vulcanized to the roller.

2.4.4 Manufacturing Costs

The molding process described in 2.4.3 is suitable for either large or small production. The estimated costs to produce 10, 100 and 10,000 units are shown in Table 2-10.



STUD. 1/4" DIA, THREADED
WELD TO WEAR PLATE
6 REQD EQ. SP.
WEAR PLATE.
1/4" THK COLD
ROLLED STEEL.
MOLD IN PLACE
1.95
1.56
1.56
20° TYP
TIRE
RUBBER, TYPE II, SPEC. MIL-T-3100.
BOND IN PLACE
2°
.55
.50 R
WHEEL
COMPRESSION MOLDING
MATL: E-260 H GLASS
FIBER MOLDING COMPOUND
(FIBERITE CORPORATION)
.95
.40
13.50 DIA
12.00 DIA
9.10 DIA
7.20 DIA
6.00 DIA
5°
4.20
OPTIONAL METHOD OF
ATTACHING WEAR PLATE



NOTES:

1. DRAWING IS FULL SIZE. SCALE DRAWING FOR MISSING DIMENSIONS.
2. MATERIAL NET WEIGHTS:
   E-260 H GLASS BMC .... 8 LBS
   (BULK FACTOR ... 8-10)
   RUBBER .... 4 LBS
   STEEL RING .... 3.8 LBS
3. THIS DRAWING HAS BEEN PREPARED FOR COST ESTIMATING PURPOSES ONLY.
4. A TRACK SUPPORT ROLLER ASSEMBLY CONSISTS OF TWO WHEEL ASSY'S MOUNTED BACK TO BACK ON A HUB.

Figure 2-21

Support Roller, Composite Material

PART NO	PART NAME	
DRAFT R ANDERSON 2-12-73		Riggs Engineering Corp.
CHECK		
DESIGN		
STRESS		WHEEL ASSY - COMPOSITE MATERIAL (AMMRC)
PROCESS		
TOOLING		
APPROVED		
		SIZE 2 / CODE IDENT NO / DRAWING NO 5072
		SCALE 1:1 / CALC. WT. / SHEET 1 OF 1

5072

2

Table 2-10 Support Roller Manufacturing Costs

<u>Support Roller</u> (Figure 2-19)

Development Engineering and Manufacturing Drawings $1500.

Quantity	10	100	10,000*
	Cost/Unit, $		
Material	111	92	58
Fabrication and Quality Control	329	277	169
Prod. Engineering and Program Adm.	114	83	10
Tooling	449	55	2
Total:	1003	507	239

*Fabrication Rate: 1000 units/year

2.5 Track Idler Wheel and Road Wheel

2.5.1 Baseline Requirements

2.5.1.1 Description

The idler wheel is located at the forward end of the track loop above ground level. It functions as a track tensioner through a linkage system connected to one of the road wheel swing arms. Although not in contact with the ground the idler wheel is subjected to very large front end impact loads. The wheel assembly consists of two identical halves mounted back to back on a hub. A solid rubber tire is vulcanized to the rim of each wheel half. The idler wheel is interchangeable with the road wheel.

The vehicle is supported by six pairs of road wheels. Since they must be interchangeable with the idler wheel they are identical to the idler wheel. The road wheels are mounted on swing arms to which the torsion bar in Paragraph 2.2 is mounted.

2.5.1.2 Dimensional Requirements

The dimensions of the wheel are given by drawings:

10887253	Wheel Assembly
10887252	Wheel, Rubber Tire
10887251	Wheel
11659122	Plate, Wear

The diameter of the assembled wheel is 26 in., including the solid rubber tire. The overall width is 7.5 in. The idler or road wheel assembly consists of two identical wheels which are mounted back to back on the hub of the swing arm assembly by means of ten 7/8" dia. bolts.

The current wheel is made from an aluminum alloy forging 2014-T6 with a 55,000 psi yield strength.

The wear plate is made from surface hardened steel.

The tire is molded from rubber Type I per Spec. MIL-T-3100 B, durometer 75±3, and is vulcanized to the wheel aluminum forging. The new design requires that the tire be replaceable.

2.5.1.3 Design Load Condition

The loads are listed in Appendix D, Table I, of the Specification and Requirements, separately for the (2) idler wheel pairs and the (12) road wheel pairs. The design loads are:

Idler Wheel:

Static Track Tension	12,000 lbs.
Static Bearing Load	24,000 lbs.
Impact Load	180,000 lbs. (Ultimate)*

Road Wheel:

Static Radial Load	4,075 to 11,250 lbs.	
Maximum Dyanmic Load	15 g	
Maximum Radial Load	73,000 lbs.	(Ultimate)*
Maximum Side Load at Wear Plate	30,000 lbs.	(Ultimate)*
Maximum Lateral Deflection	.5 inch	

*These loads have been established as ultimate loads by the Contracting Officer's Representative, Mr. J. Plumer. Other loads listed are limit loads. A safety factor of two is used for limit loads.

To estimate the endurance requirements it is calculated that at a maximum vehicle speed of 30 mph, the rpm of the wheel is,

$$N = \frac{10,084}{26} = 388 \text{ rpm}$$

This gives an approximation of the radial load change from minimum to maximum at a frequency of 388/60 = 6.5 per sec. Assuming a test mileage requirements of 3,000 miles at 30 mph, the minimum life expectancy would be 3,000/30 = 100 hrs. This is equivalent to $388 \cdot 60 \cdot 100 = 2.33 \cdot 10^6$ load cycles.

The design loads in Table I of Appendix D of the Specifications and Requirements document are referenced to the drawing of the wheel assembly and it is therefore assumed that the loads are acting on the two wheel halves shown on Drawing No. 10887253. Consequently the half wheel assembly is subjected to 50% of these loads, with the exception of the side load all of which is reacted by one wheel half.

The design limit loads for the idler and the road wheel halves are summarized in Figure 2-22 . It shows that the largest radial load is the front impact load combined with track tension load.

The radial component of the track tension load is distributed approximately parabolic along the periphery of the wheel. The relative magnitude at a track plate segment was shown in Figure 2-13 for the drive wheel and is about 30% of the resultant bearing load caused by track tension. At the specified static bearing load of 12,000 lbs. per half wheel, the maximum load at the track plate is $12,000 \cdot .30 = 3,600$ lbs. This combined with the forward impact load of 90,000 lbs. results in a maximum radial load for the idler wheel of:

$$P = 90,000 + 36,000 \cdot \cos 13^\circ = 93,508 \text{ lbs.}$$

This load is assumed to be an occasional one and requires a design safety factor of greater than one for ultimate material strength.

The maximum radial load of 36,500 on the road wheel half is also considered as an occasional load and would require a safety factor of greater than one for ultimate material strength.

The road load of 11,250 lbs. is, however, a reoccuring load and the design should be capable of at least $2.33 \cdot 10^6$ load cycles. The R-factor can be estimated from the quoted minimum and maximum road wheel loads: $R = 4,075/11,250 = .36$. This value can be used to design a Goodman diagram and to determine design allowables for $2.33 \cdot 10^6$ load cycles. For this load condition the safety factor is one for allowable material fatigue strength.

The side load of 30,000 lbs. is considered as an occasional load to which a safety factor of greater than one for ultimate material strength should be applied. This load may act simultaneously with the radial load of 36,500 lbs.

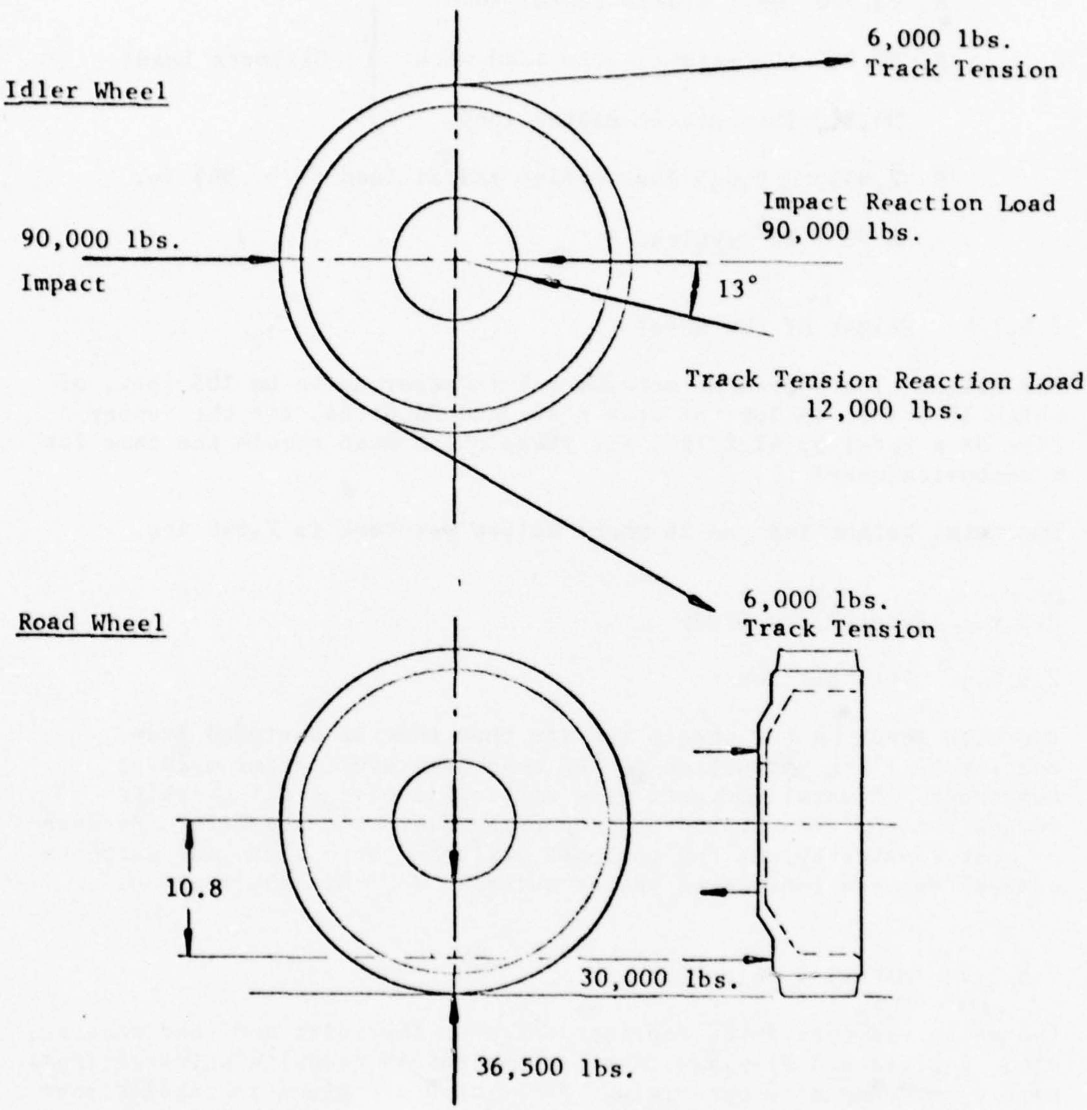


Figure 2-22 Design Loads for Idler and Road Wheels

The composite wheel will be designed for the following three load conditions.

$$\left.\begin{array}{l}\text{A. 93,500 lbs. static radial load} \\ \text{B. 30,000 lbs. static side load with} \\ \quad\text{36,500 lbs. static radial load}\end{array}\right\} \text{Ultimate Loads}$$

C. 2,037 to 5,625 lbs cyclic radial load (R = .36) for

$2.33 \cdot 10^6$ cycles.

2.5.1.4 Weight of the Wheel

The weight of the present metal wheel is reported to be 105 lbs., of which 15.6 lbs. is for the wear plate and 26.6 lbs. for the rubber tire or a total of 42.2 lbs. for items which must remain the same for a composite wheel.

The total weight for the 28 wheel halves per tank is 2,940 lbs.

2.5.2. Feasibility Study

2.5.2.1 Introduction

The high loads on the wheels require that they be designed from continuous fiber composites rather than from short fiber molding compounds. Several concepts were studied involving all graphite composites and also hybrid designs using glass and graphite. Because of cost considerations the concepts utilizing both glass and graphite composites were considered better suitable for this application.

2.5.2.2 Material Selection

The materials considered for the design of the idler and road wheels are: E-glass and S2-glass fabric/epoxy and AS graphite unidirectional tape/epoxy composite materials. Properties are given in tabular form in previous paragraphs but are repeated in this analysis as they appear in the formulas.

2.5.2.3 Stress Analysis

Application of the radial load by the track plate causes a flattening of the rubber tire. The contact width "b" is calculated from:

$$b = 2.154 \sqrt{\frac{Q \cdot R_o}{L} \left(\frac{1}{E_1} + \frac{1}{E_2} \right)}$$

Where

R_o = Tire outer radius = 13 in.

L = Loaded tire width = 5 in.

E_1 = Modulus of the track plate. Steel = $29 \cdot 10^6$ psi

E_2 = Modulus of the rubber tire with a durometer hardness of 75 = 2,300 psi

Q = Radial load

Then for a maximum static load of 5,625 lbs. (Load Condition "C")

b = 8.6 in.

The maximum parabolic contact pressure is:

$$P_{max} = .592 \sqrt{\frac{Q}{L \cdot R_o} \Big/ \left(\frac{1}{E_1} + \frac{1}{E_2} \right)}$$

or at maximum static load:

P_{max} = 275 psi

For dynamic and impact loads, which are applied instantaneously, the solid rubber has no time to deflect and is considered as almost rigid. However for the analysis it is assumed that the above maximum static deformation is already present. Consequently the radial load is not concentrated, but distributed over a chord length of 8.6 in. The parabolic pressure distribution has been replaced by two concentrated loads located at $.1875 \cdot b = .1875 \cdot 8.6 = 1.6$ in. from the maximum pressure point.

With reference to the wheel width, it is assumed that the load is transmitted uniformly through the tire to the full 6 in. width of the wheel rim.

For a one inch wide ring of the rim, the design limit loads for Condition B (road wheel) is then 36,500/6 = 6,084 lbs/in, represented by two loads of 3,042 lbs. each. The reaction to these loads is a shear flow at the sides of the ring element.

The radial load and the shear flow cause bending in the wheel rim. The stresses can be calculated if the radial thickness of the rim is known. In order to carry the load across the wheel rim uniformly, it is required that the radial deflection of the rim across its width be as uniform as possible. It is obvious, that this depends on the effective stiffness of the rim ring segments. The maximum stiffness is at the disc of the wheel. A rim of constant thickness would deflect radially more at the unsupported outer ring segment than at the disc. The deflection can be controlled by providing:

A. Thicker rim ring element at the unsupported end of the rim.

B. Gussets to support the outer ring elements.

C. Increasing the effective material modulus of elasticity towards the outer ring element.

Figure 2- 23 shows one concept of a composite material design of the wheel. The wheel is molded from segments cut from woven fabric which form the disc and rim. A tapered internal doubler disc is bonded to the basic disc. A ring consisting of circumferentially wound fibers is bonded to the inside of the unsupported edge of the rim. The rubber tire is vulcanized to an aluminum ring which is bolted to the wheel. This arrangement makes the rubber tire replaceable. The bolts also hold the wear plates in place.

For a preliminary analysis the rim is divided into three ring sections of 2 in. width each. Each ring is loaded by two concentrated forces $2 \cdot 3{,}042 = 6{,}048$ lbs. each (Condition B). The reaction is a shear flow at the sides of the ring element. For the analysis it is assumed that the rubber tire and the aluminum ring do not contribute to the bending strength of the composite wheel structure.

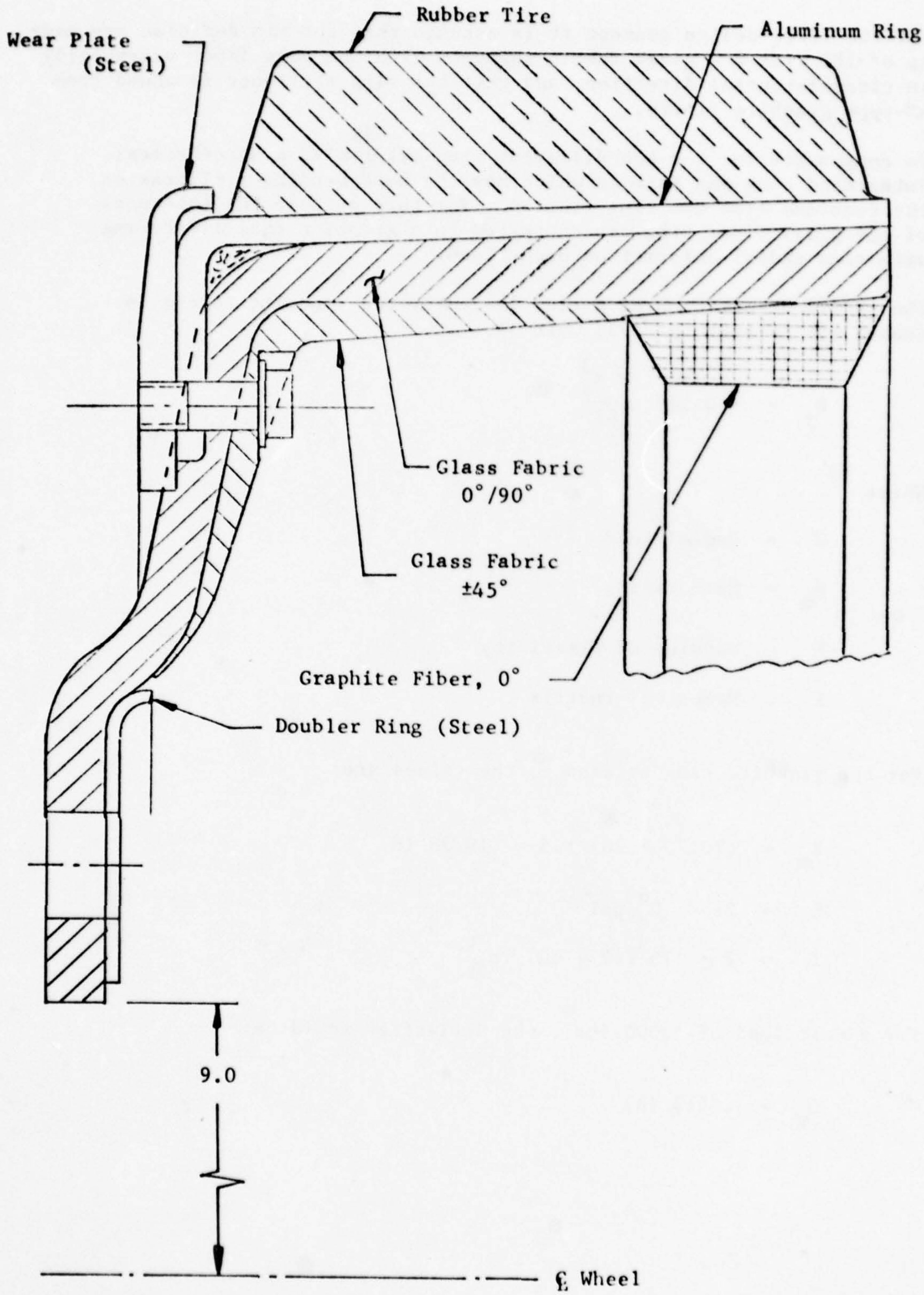


Figure 2-23 Idler Wheel Design Concept

For the first design concept it is assumed that the rim and disc are made up of 181 type S-2 glass fabric segments with the warp fiber essentially in circumferential direction, and that the ring stiffener is wound from AS-type graphite fibers.

To compensate for the two different ring materials, a hypothetical integrated ring was assumed which has the same bending stiffness as the combined disc and ring section. For this purpose the thickness of the glass fiber ring was converted to a graphite equivalent ring with same radial deformation under load.

The radial deformation of a ring loaded by two opposite forces is (Roark 4th Edition, p. 172, Case 1):

$$D_y = -0.149 \frac{W R_m^3}{EI}$$

Where

W = Radial load

R_m = Mean radius

E = Modulus of elasticity

I = Moment of inertia

For the graphite ring stiffener, the values are:

$R_m = (10.75 + 10) \cdot .5 = 10.38$ in.

$E = 21 \cdot 10^6$ psi

$I = 2 \cdot .75^3/12 = .07$ in.^{4}

For a unit load of 1,000 lbs., the deflection would be:

$D_y = .0113$ in.

For the glass fabric ring section, the values are:

$$R_m = (11.43 + 10.75)\ .5 = 11.09 \text{ in.}$$

$$E = 4 \cdot 10^6 \text{ psi}$$

$$I = 2 \cdot .68^3/12 = .0524 \text{ in.}^4$$

In order to deform the glass fabric ring as much as the graphite ring, the partial load on the glass ring would be:

$$W = .0113 \frac{4 \cdot 10^6 \cdot .0524}{.149 \cdot 11.09^3} = 11.65 \text{ lbs.}$$

The thickness of graphite replacement ring would then be:

$$\frac{I}{R^3} = \frac{.149 \cdot 11.65}{21 \cdot 10^6 \cdot .113} = .00000732$$

This value is fulfilled when the ring thickness is .40 in.

Consequently the total thickness of the hypothetical graphite fiber ring is .75 + .40 = 1.15 in. with a mean radius of 10 + .58 = 10.58 in.

The load diagram for the hypothetical graphite ring, is shown in Figure 2- 24 (Load Condition B).

1. <u>Shear Flow Analysis</u>:

$$S = \frac{Q}{\pi R_m} \cdot \sin\alpha \text{ (lb/in)}$$

Where

Q = Load on the rim ring

R_m = Mean radius of the rim ring

α = Angle between 0° and 180° counting from the load contact point

$S = 366 \cdot \sin\alpha$ lb/in

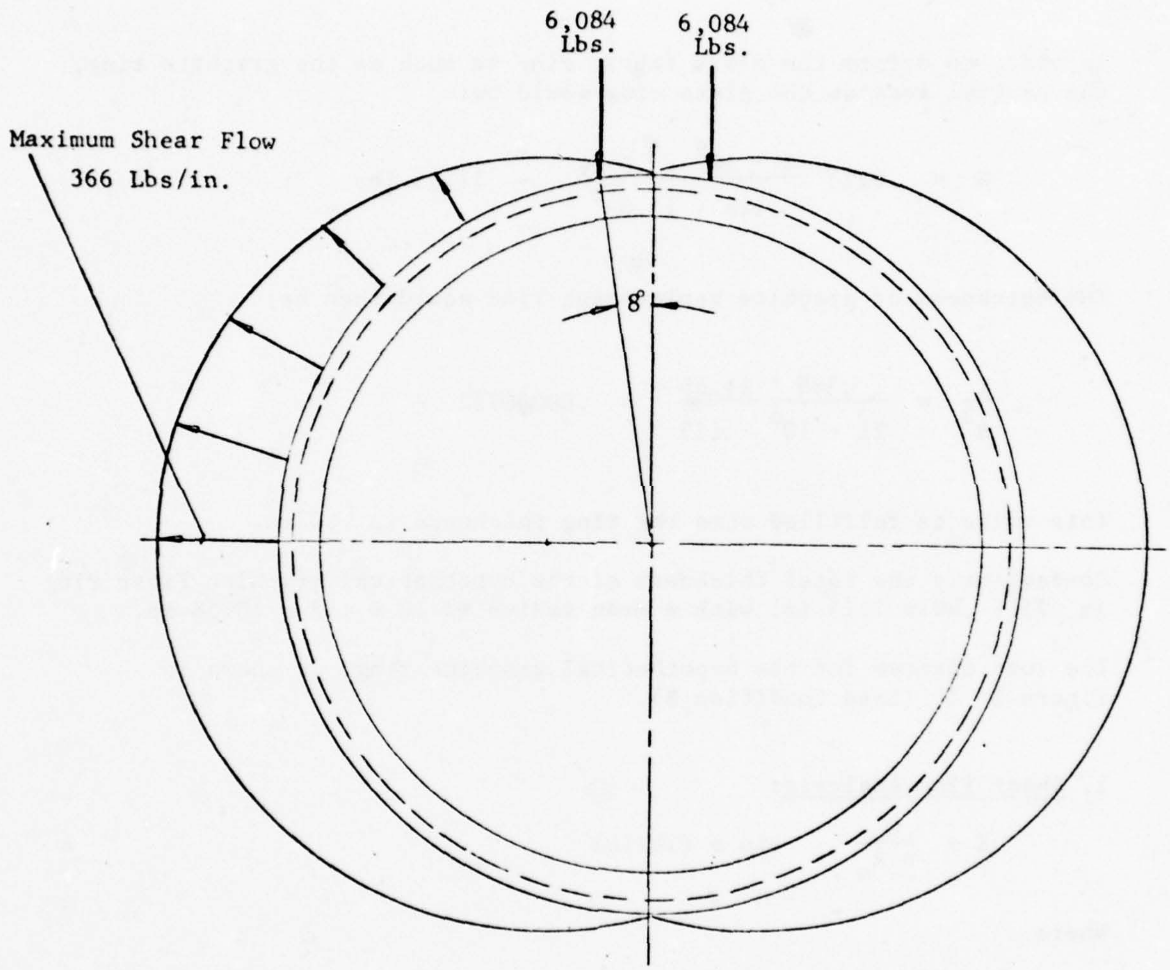


Figure 2-24 Load Distribution on a 2 in. Wide Ring (Cond. B)

α°	$\sin \alpha$	S lb/in/2 in. Rim
0	0	0
30	.500	183
60	.866	317
90	1.000	366

Therefore the maximum shear flow is 366 lb/in. Since the hypothetical ring is actually supported by a glass fabric composite of .75 in. thickness, the shear stress in the composite (0°/90° orientation) is:

$$f^s = \frac{366}{.75} = 488 \text{ psi}$$

At the disc, into which the entire radial load is to be transferred, the maximum shear stress is approximately:

$$f^s = \frac{366}{.90} \cdot 3 = 1,220 \text{ psi}$$

The ultimate in-plane shear stress in a 0°/90° E-glass fabric is 16,000 psi with the proportional limit of about 2,500 psi (MIL-HDBK-17A; 4-65). At ±45° the in-plane ultimate shear is 34,000 psi with a proportional limit of about 15,000 psi. For load Condition B the applied limit stress is less than the allowable proportional stress. The safety factor for ultimate stress is high.

For load Condition A, the applied stress would be 93,500/36,500 = 2.56 times greater:

$$f^s = 1,220 \cdot 2.56 = 3,123 \text{ psi}$$

here the safety factor for ultimate stress is also high, but the 0°/90° proportional limit would be exceeded. This can be corrected by laminating the internal doubler ring from ±45° laminated fabric.

For an estimate of stresses in the 0°/90° and the ±45° composite, the shear load is divided based on the relative shear stiffness of the laminates.

The reference value for the glass fabric are:

Orientation	0°/90° (1)	±45° (2)
In-plane shear Modulus, G	$.80 \cdot 10^6$	$1.8 \cdot 10^6$
Laminate thickness, t	.55	.35

Then

$$Q_1 = Q_2 \frac{t_1 \cdot G_1}{t_2 \cdot G_2}$$

Where

Q = Shear flow

1 = 0°/90° laminate

2 = ±45° laminate

And

$$Q_1 = .70\ Q_2$$

The total shear flow Q = 366 lb/in/2 in. rim

Then

Q_2 = 215 lb/in

Q_1 = 151 lb/in

With this load distribution the maximum shear in the rim stresses at the mounting disc are for Load Condition A:

0°/90° f_1^S = 2,109 psi

±45° f_2^S = 4,718 psi

These shear stresses are within the proportional limit and therefore acceptable. The safety factor for ultimate material strength is large.

2. <u>Bending Analysis</u> - The bending moments were calculated following the procedure described in Paragraph 2.4, Support Roller. The maximum bending moment is at $\alpha = \theta = 8^\circ$. The coefficients "K" are:

α°	α rad	Z $\sin \alpha$	U $\cos \alpha$	U^2 $\cos \alpha^2$	K_2	K_1
8	.1396	.1392	.9903	.9806	.0581	.1760

The bending moment on the hypothetical ring section is:

$$M = Q \cdot R \cdot K_1$$

Where

Q = 12,168 lbs. (for a 2" wide ring)

R = 10.58 in.

Then at $\alpha = \theta = 8^\circ$:

M = 22,658 in. lbs.

With the wall thickness of 1.15 in., the bending stress is:

f^b = 51,398 psi

The allowable bending stress for unidirectional AS-graphite is 180,000 psi. Therefore the margin of safety for Load Condition B is:

$$M.S. = \frac{180,000}{51,398} - 1 = +2.50$$

and for Load Condition A:

$$M.S. = \frac{180,000}{51,398 \cdot 2.56} -1 = +.37$$

3. Rim Analysis for Cycling Loading - The applied stresses in the rim and the ring stiffener at 5,625 lbs. radial pressure are calculated by reduction of the stresses calculated for Condition B by the load ratio 5,625/36,500 = .154.

Consequently the maximum cycling stresses are:

Shear at the mounting disc:

0°/90° Composite (glass cloth)

$$f_1^s = \frac{2,109}{2.56} \cdot .154 = 127 \text{ psi}$$

±45° Composite (glass cloth)

$$f_1^S = \frac{4,718}{2.56} \cdot .154 = 284 \text{ psi}$$

Bending in the ring section:

0°/90° Composite (glass cloth)

$$f_1^b = 21,176 \cdot .154 = 3,261 \text{ psi}$$

Unidirectional composite (graphite)

$$f_2^b = 54,210 \cdot .154 = 8,348 \text{ psi}$$

Compared with allowable material stresses at $2.33 \cdot 10^6$ load cycles, all the above stresses are very low. Therefore the design of the wheel rim is determined by the high radial impact loads on the idler wheel.

4. Analysis of the Disc - The disc by which the wheel is bolted to the flange, transfers shear and bending stresses from the rim to the flange. The ultimate loads to which the disc is subjected for Load Condition B are summarized in Figure 2-25. The vertical load of 36,500 lbs. is introduced into the plate by a pheripheral shear flow. The maximum value is 1,115 lb/in (at 10.43 in. radius). It is resisted by a bearing load on the 9 in. diameter hole of 36,500/9 = 4,056 lb/in. The vertical load also causes a bending moment of the magnitude of 36,500 · 4.25 = 155,125 in. lbs. at the mounting bolts, which is resisted by tension in the bolts and by contact pressure against the flange at the bolt circle.

The side load of 30,000 lbs. is transmitted to the mounting bolts by shear and results in tension in the bolts. The moment, 30,000 · 10.80 = 324,000 in. lbs., is resisted by tension in the bolts and by contact pressure at the bolt circle. The total bending moment which is acting on the disc and must be reacted by the bolts is 155,125 + 324,000 = 479,125 in. lbs.

5. Bending Analysis - The rim of the wheel, together with the replaceable tire is very rigid in bending. Therefore it can be assumed that the edges of the disc are fixed. The applied ultimate load of 479,125 in. lbs. is equivalent to a moment acting on a trunnion of a diameter somewhat greater than the bolt circle, say 13 in.

For definition of bending stresses several published methods were investigated and the results compared. These methods are reported in:

- Timoshenko, Plates and Shells, 2nd Edition, p. 289
- Roark, Formulas for Stress and Strain, 4th Edition, Table X, p. 217 & 219
- Roark, Formulas for Stress and Strain, 4th Edition, p. 242

The Timoshenko formulas for fixed edge condition include a Poisson's ratio of .30 which applied to metals, but is not correct when used for composites.

The Roark Table X formula has provisions for different Poisson's ratios, however the application for fixed edges condition is limited to relatively small trunnion diameters.

The Roark p. 242 equations are based on coefficients which were developed for a Poisson's ratio of .30, however stresses for fixed edges as well as for supported condition can be estimated.

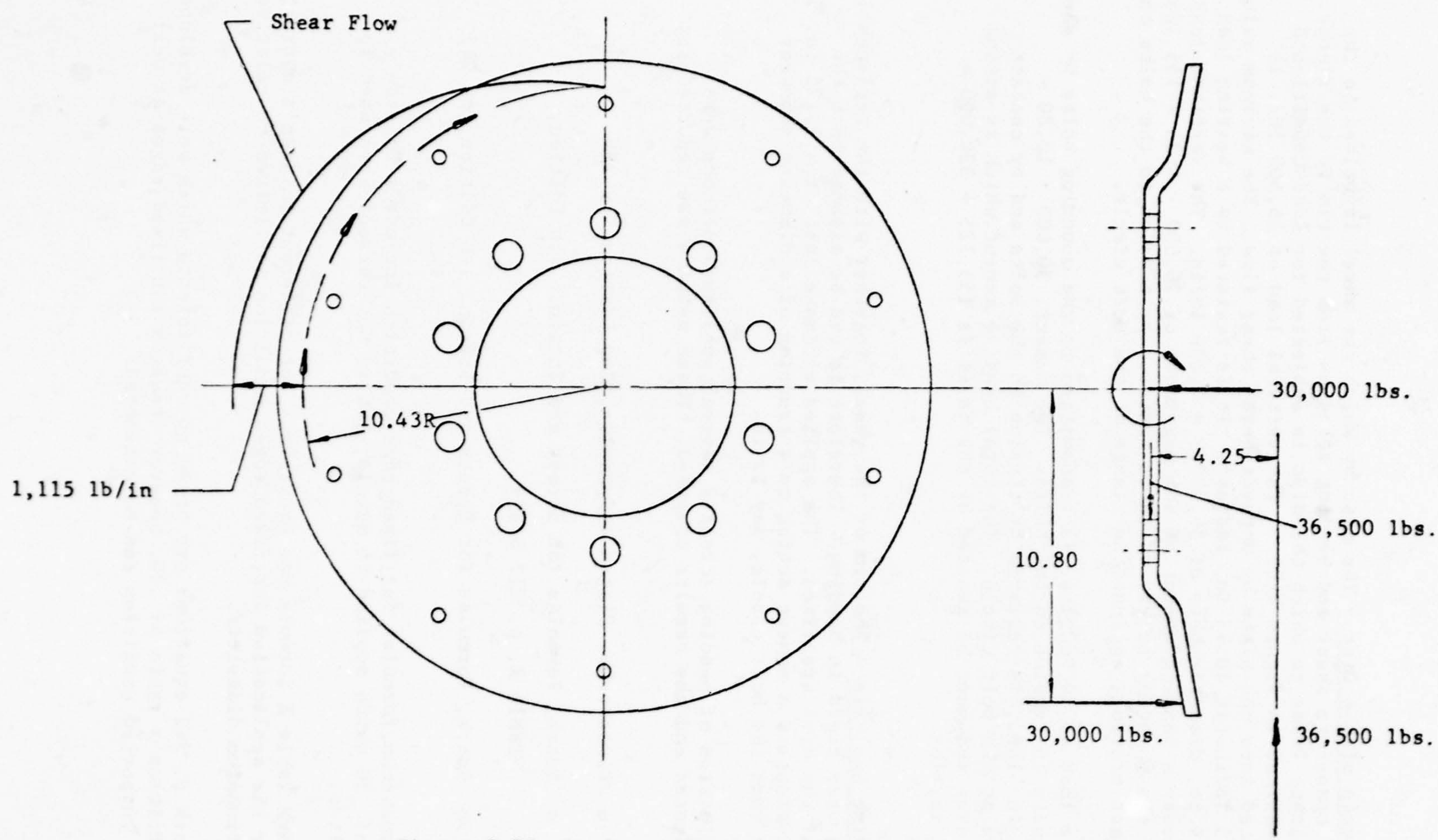


Figure 2-25 Loads on Wheel Disc (Cond. B)

The results of calculating with the different formulas are presented in the following. Only the maximum bending stress at the radius of 6.5 in. is listed. The calculated stresses are based on the following disc dimensions:

$$r_i = 6.5 \text{ in.}$$

$$r_o = 10.43 \text{ in.}$$

$$t = .52 \text{ in.}$$

$$r_i/r_o = .62 \text{ in.}$$

Timoshenko, p. 289

Fixed edges, $\nu = .30$

$$f^b_{r_i} = \frac{\alpha}{\alpha_2} \frac{M}{r_o \quad t^2}$$

Where for $r_i/r_o = .62$

$$\alpha = 20.15$$

$$\alpha_2 = 30.50$$

$$f^b_{r_i} = 112{,}125 \text{ psi}$$

Per Roark, Table X, Case No. 5

supported edges $\nu = .30 \quad m = 1/.30 = 3.33$

$$f^b_{r_i} = \frac{3M}{4 \cdot \pi \cdot t^2 \cdot r_1} \left[1 + \left(\frac{m+1}{m}\right) \cdot \log \frac{2\,(r_o - r_i)}{K \cdot r_o} \right]$$

Where

$$K = \frac{.49 \cdot r_o^2}{(r_i + .70 \cdot r_o)^2} = .27986$$

Then

$$f^b_{r_i} = 148{,}886 \text{ psi}$$

Same, but $\nu = .12 \quad m = 8.33$

$$f^b_{r_i} = 137{,}186 \text{ psi}$$

Per Roark p. 242

$\nu = .30$

The bending stress at the radius of 6.5 in. is given by:

$$f^b = \beta \frac{M}{r_o \cdot t^2}$$

Where β depends on the edge conditions and on the r_i/r_o ratio, in this case = .62.

	β
Edges supported	.997
Edges fixed	.688

(Compared with .660 by Timoshenko)

The maximum stresses for the supported edges are:

$$f^b_{r_i} = 169{,}377 \text{ psi}$$

The maximum stresses for the fixed edges are:

$$f^b_{r_i} = 116{,}882 \text{ psi}$$

The following summary shows the maximum bending stresses, in psi, at the trunnion for different methods of analysis (ν = .12):

	Supported	Fixed
Timoshenko	-	103,155
Roark, Table X	137,186	-
Roark, p. 242	155,827	107,531
Average	146,506	105,343

This shows that the stresses for the fixed edge condition are smaller (about 28%) than the supported edge condition. Because the design uses continuous fabric layers in the corner between disc and rim the fixed edge condition is justified.

However the actual thickness of the disc is not uniform, but tapered (Figure 2-23). This causes less deflection and therefore would result in less bending stress than the constant thickness disc calculated above.

Prior to estimating the stresses in the tapered disc, the ±45° doubler must be converted numerically into a 0°/90° doubler, so that a uniform modulus of elasticity can be used.

The conversion is carried out by making the bending parameter Et^3 for the actual $\pm 45^\circ$ laminate and the hypothetical $0^\circ/90^\circ$ laminate equal. This results in the $0^\circ/90^\circ$ doubler with the maximum thickness of:

$$t_{0^\circ/90^\circ} = \sqrt[3]{\frac{(Et^3)_{45^\circ}}{E_{0^\circ/90^\circ}}}$$

Where

$$E_{\pm 45^\circ} = 2.5 \cdot 10^6 \text{ psi}$$

$$E_{0^\circ/90^\circ} = 4 \cdot 10^6 \text{ psi}$$

$$t_{\pm 45^\circ} = .40 \text{ in.}$$

Then

$$t_{0^\circ/90^\circ} = .342 \text{ in.}$$

The analysis is now carried out according to Timoshenko, Theory of Plates and Shells, 2nd Edition, p. 303. This, however shows a method to calculate stress, when the tapered disc is loaded by central force and not by a bending moment. For bending of a tapered disc, no methods of analysis are given. Therefore an approximate analysis is carried out by comparing coefficients of a centrally loaded tapered disc with a centrally loaded constant t disc and with a bending moment loaded constant t disc as already calculated. The side load of 30,000 lbs. is selected as the center load. For the comparative analysis $\nu = .30$ was used.

The radius ratio is $r_o/r_i = 1.6$. According to Timoshenko the maximum stresses and deflections are:

flat disc $\quad f^b = K \dfrac{P}{h^2} \qquad d_i = K_1 \dfrac{P \cdot r_o^2}{E \cdot h^3}$

tapered disc $\quad f^b = K' \dfrac{P}{h_1^2} \qquad d_1' = K_1' \dfrac{P \cdot r_o^2}{E \cdot h_1^3}$

Where K - Coefficient (extrapolated for r_o/r_i = 1.6)

$K = .257$

$K' = .0099$

$K_1 = .535$

$K_1' = .0211$

$h = .52$ in.

$h_1 = .862$ in.

Then the estimated maximum stresses are:

flat disc

$f^b = 28,513$ psi

tapered disc

$f^b = 21,600$ psi

To compensate for the lower Poisson's ratio, the stresses are reduced by 8% as in the previous case,

flat disc

$f^b = 22,810$ psi

tapered disc

$f^b = 17,280$ psi

This shows that the maximum bending stress for the tapered disc is only 76% of the stress for a flat disc. Consequently the previously calculated bending stress in the .52 in. flat disc of 105,343 psi can be reduced for a tapered disc per Figure 2-23 to 105,343 · .76 = 80,061 psi.

Since the side load of 30,000 lbs. must be applied as a center load as well as a bending moment, the resultant bending stress in the tapered disc will be:

$$f^b_{res} = 97,341 \text{ psi}$$

Then the margin of safety for ultimate material strength is:

$$M.S. = \frac{113,500}{97,341} - 1 = +.17$$

6. Estimate of Lateral Deflections - The bending moment causes a tilting in the flat disc of (Timoshenko p. 289):

$$\varphi = \frac{M}{\alpha_2 \cdot E \cdot h^3}$$

$$\varphi = 1.6°$$

At a radius of 13 in. this is equivalent to .363 in. lateral deformation.

The lateral deflection caused by the load in the center of the disc,

Flat disc

$$d_1 = .057 \text{ in.}$$

Tapered disc

$$d_1' = .027 \text{ in.}$$

or 47% of the flat disc

Assuming the tilting deflection of a tapered disc is also reduced by the same factor, the total lateral deflection of the wheel is then:

$$d = (d_2 \cdot .47) + d_1'$$

$$d = .198 \text{ in.}$$

Since the maximum allowable deflection is .50 in., the lateral stiffness of the wheel is more than adequate.

7. <u>Shear Stresses</u> - The shear stresses in the plate caused by the lateral force of 30,000 lbs. is a maximum at the bolt circle and is composed of the uniform lateral shear and the non-uniform bending shear at the reference radius of 6.5 in.

The lateral shear stress is:

$$f_L^S = \frac{30,000}{2 \cdot \pi \cdot 6.5 \cdot .52} = 1,413 \text{ psi}$$

The bending shear has an arc shaped distribution. The maximum shear stress is estimated from the section modulus of the critical disc shear section.

$$Z = \pi \cdot 13^3/32 = 216 \text{ in.}^3$$

Then the maximum shear stress due to bending is:

$$f_B^S = \frac{30,000 \cdot 10.8}{216} = 1,500 \text{ psi}$$

The maximum total shear stress is at the lower half of the wheel disc:

$$f_{max}^s = 1,413 + 1,500 = 2,913 \text{ psi}$$

The allowable ultimate shear stresses for a $0°/90°$ glass fabric laminate are:

Rail shear	=	16,000 psi
Interlaminar Shear	=	8,350 psi

which means that the margin of safety is large.

The analysis has shown that the critical stresses in the disc are bending stresses due to eccentricity of the load. However all applied stresses have a safety factor greater than "one" for ultimate material strength and therefore the design concept shown in Figure 2-23 is acceptable.

2.5.2.4 Weight

	lb.	%
Composite Wheel	46.2	45
Tire Support Ring (Aluminum)	12.1	12
Tire (Rubber)	26.5	26
Wear Plate (Steel)	15.7	15
Doubler Ring (Steel)	2.2	2
Total:	102.7	100

A comparison of the weights shows that the composite wheel does not have a significant advantage over the aluminum wheel which weighs 105 lbs. The aluminum forging alone was estimated to 62.7 lbs. The equivalent composite part is 46.2 lbs. This is a weight saving of 16.55 lbs. or 26%. However the new requirement of a removable tire and the need for the doubler to reinforce the mounting holes add 14.3 lbs. practically eliminating the weight saving by the composite material.

Since the wheel is designed for impact loads and not for the cyclic loads, the impact load on the idler wheel is critical. This load is 2.5 times greater than the impact load on the road wheel. Weight saving could be achieved by eliminating the requirement for interchangeability and design the idler wheel and the road wheel for their load requirements.

2.5.3 Manufacturing Analysis

A drawing of the idler wheel and road wheel is shown on Figure 2-26. The disc and rim consists of layed up segments of glass fiber fabric in a pattern shown on the drawing. Material for both -1 and -2 are layed up on a mandrel which after compaction and prebleed is inserted in a female mold and cured. Layup on a male tool is easier than working the material into a cavity and in this case where it is desirable to control the outside contours curing in a female tool is better. Pressure can be applied either by vacuum bag, which should be of a reuseable and contour molded type or by thermo elastomeric tooling where the male mandrel would have a rubber jacket of sufficient thickness to provide pressure on the laminate due to thermal expansion. The stiffener ring, (item 3 on Figure 2-26) consists of wound unidirectional graphite fiber tape and is cured separately. The ring is bonded to the rim. The mating surfaces are machined slightly conical to assure good contact pressure during bonding.

The rubber tire, which must be of a replaceable type, is vulcanized to an aluminum ring which fits over the composite wheel rim. It is mounted to the wheel by ten bolts. These bolts are also the attachments for the steel wear plates, which, except for the hole pattern, are identical to the wear plates on the existing metal wheel.

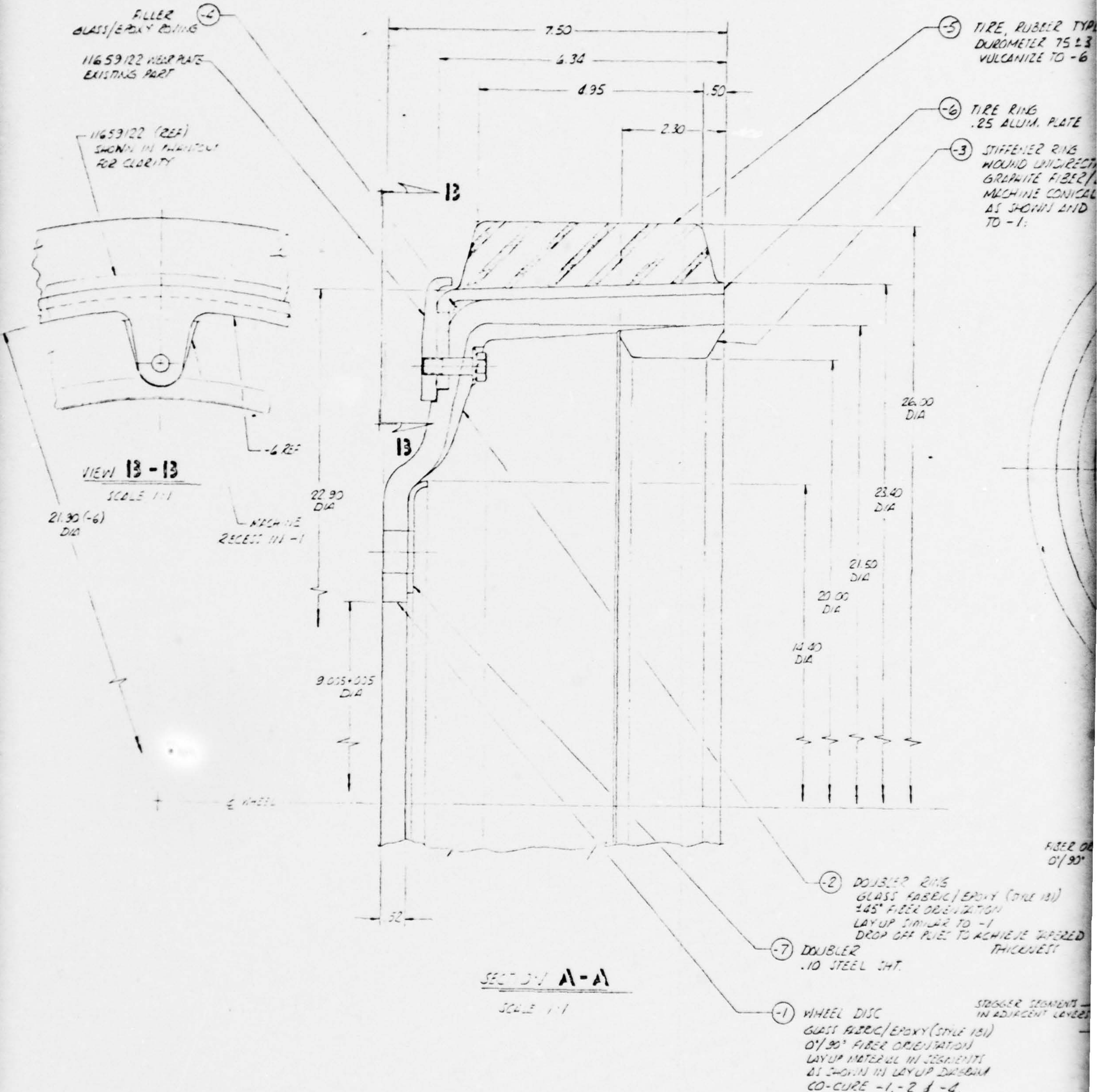
The wheel halves are mounted back to back on a hub by ten 7/8 inch diameter bolts. To prevent damage to the composite surface during wrenching a doubler ring has been added to the inside surface of the wheel disc.

2.5.4 Manufacturing Costs

The design of the idler and road wheel shown in Figure 2-26 is suitable for production in quantities less than 500. As pointed out in the discussion of the drive wheel in paragraph 2.3.4 for large quantity production further studies in fabrication techniques must be conducted and new low cost methods must be developed in order to produce a cost effective part.

Estimated costs to manufacture the idler and road wheel are shown in Table 2-11 for quantities of 10, 100 and 10,000 units.

Costs could be reduced for the road wheel if the requirement for interchangeability with the idler wheel were eliminated. The load requirements for the idler wheel are much more severe than for the road wheel.



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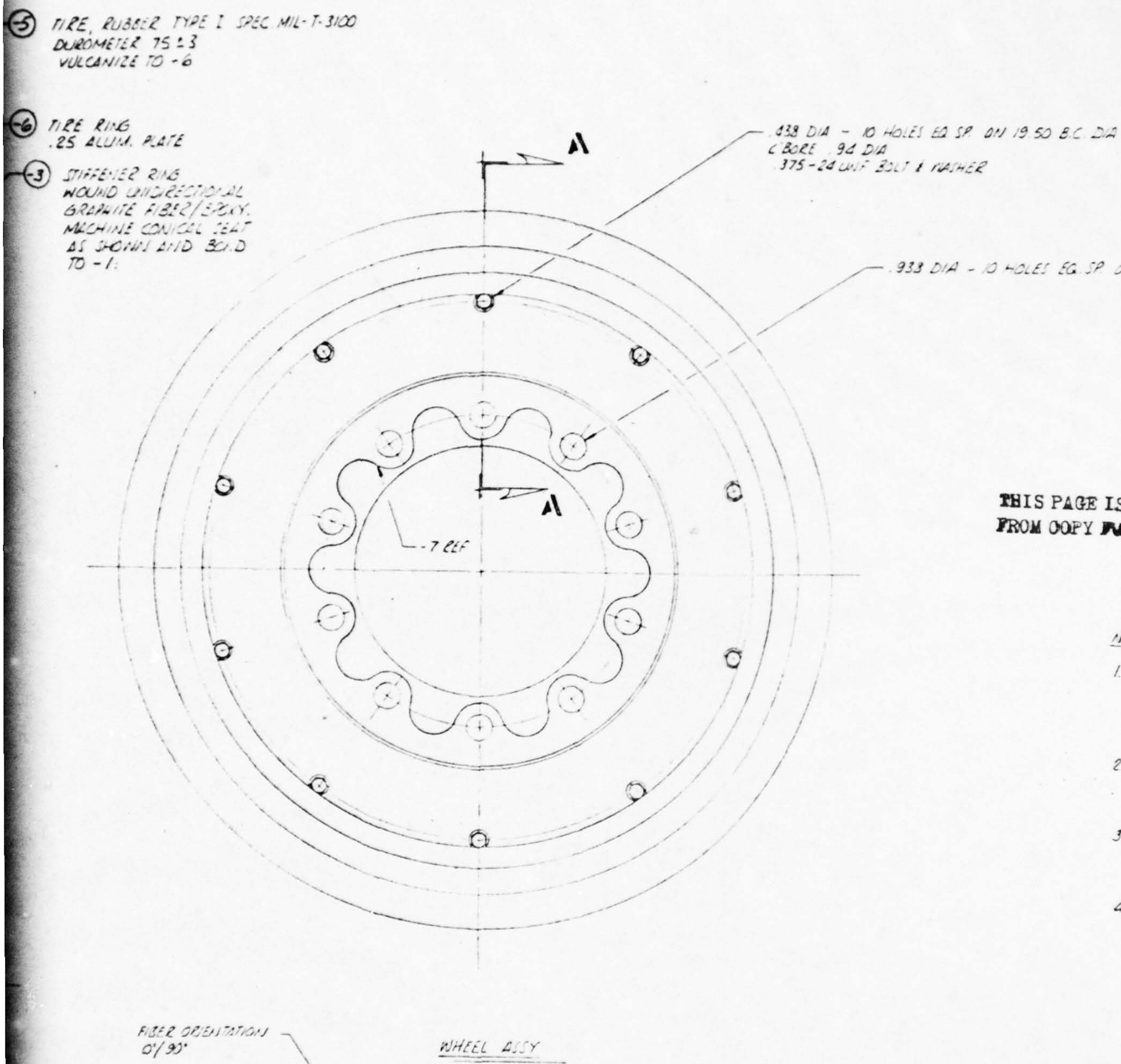

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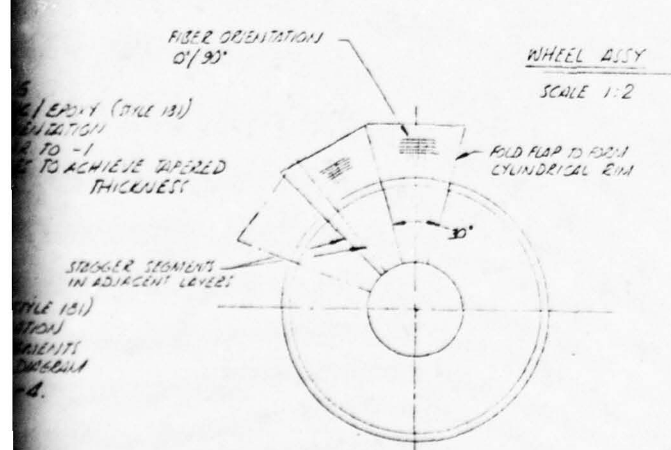
NOTES:

1. THIS DRAWING HAS BEEN PREPA ESTIMATING PURPOSES ONLY AN DIMENSIONED. SCALE DRA DIMENSIONS.
2. MATERIAL: GLASS FIBER FABRI AND GRAPHITE F EPOXY.
3. COMPOSITE MATERIAL NET WEIG GLASS FABRIC/EPOX GRAPHITE TAPE/EPO
4. WEIGHT OF OTHER ITEMS:
 - 5 TIRE (RUBBE
 - 6 RING (ALUM)
 - 7 DOUBLER (STL 11659122 WEAR PL



LAY UP DIAGRAM OF -1

LAY UP OF -2 IS SIMILAR EXCEPT FOR FIBER ORIENTATION (± 45°)

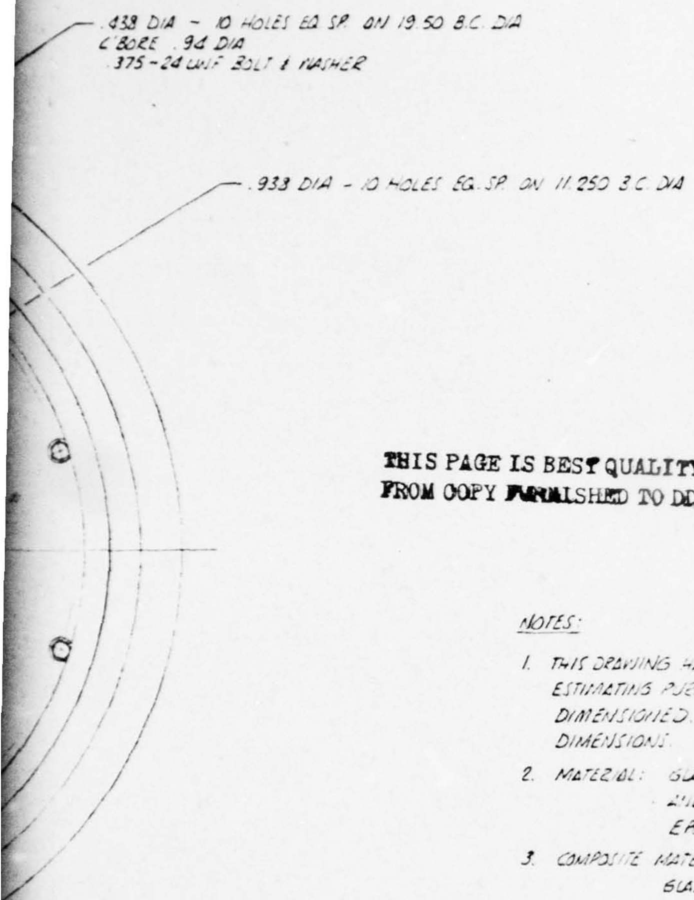
2

Figure 2-26

Idler & Road Wheel

Composite Mate

QUANTITY REQUIRED	PART NO	PART NAME	
MANUFACTURING TOLERANCES ARE UNLESS OTHERWISE NOTED: DECIMALS .X ± .1 .XX ± .03 .XXX ± .010 ANGLES ± 1/2 DEG. ALL MACHINED SURFACES ARE AS NOTED	DRAFT		Riggs Engineerin
	CHECK		
	DESIGN		WHEEL ASSY
	STRESS		COMPOSITE
	PROCESS		
	TOOLING		
	APPROVED		
NEXT ASSEMBLY / NO. REQ'D			SIZE R / CODE IDENT NO
CONTRACT NO			SCALE NOTED



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NOTES:

1. THIS DRAWING HAS BEEN PREPARED FOR COST ESTIMATING PURPOSES ONLY AND IS NOT FULLY DIMENSIONED. SCALE DRAWING FOR MISSING DIMENSIONS.
2. MATERIAL: GLASS FIBER FABRIC (STYLE 131)/EPOXY AND GRAPHITE FIBER ROVING OR TAPE/EPOXY.
3. COMPOSITE MATERIAL NET WEIGHTS:
   - GLASS FABRIC/EPOXY .... 40.6 LBS
   - GRAPHITE TAPE/EPOXY ..... 5.6 LBS
4. WEIGHT OF OTHER ITEMS:
   - -5 TIRE (RUBBER) .... 26.6 LBS
   - -6 RING (ALUM) .... 12.1 LBS
   - -7 DOUBLER (STL) ...... 2.2 LBS
   - 11659122 WEAR PLATE (STL) .... 15.7 LBS

5076

Figure 2-26

Idler & Road Wheel

Composite Material

QUANTITY REQUIRED	PART NO	PART NAME
MANUFACTURING TOLERANCES ARE UNLESS OTHERWISE NOTED: DECIMALS .X ± .1, .XX ± .03, .XXX ± .010; ANGLES ±1/2 DEG. ALL MACHINED SURFACES ARE AS NOTED	DRAFT 5-10-78	Riggs ENGINEERING CORP. SAN DIEGO, CALIFORNIA
	CHECK	WHEEL ASSY, IDLER & ROAD WHEEL COMPOSITE MAT'L (AMMRC)
	DESIGN	
	STRESS	
	PROCESS	
	TOOLING	
	APPROVED	
NEXT ASSEMBLY / NO REQ'D		SIZE R / CODE IDENT NO / DRAWING NO 5076
CONTRACT NO		SCALE NOTED / CALC WT / SHEET 1 OF 1

3

Table 2-11 Idler and Road Wheel Manufacturing Costs

Idler and Road Wheel (Figure 2-26)

Development Engineering and Manufacturing Drawings $2500.

Quanitity	10	100	10,000*
	Cost/Unit, $		
Material	1231	1015	699
Fabrication and Quality Control	665	600	341
Prod. Engineering and Program Adm.	270	207	20
Tooling	1288	209	8
Total:	3454	2031	1068

*Fabrication Rate: 1000 units/year

2.6 Track End Connector Link

2.6.1 Baseline Requirements

2.6.1.1 Description

The track end connector links tie the track shoe assemblies together to form a continuous track. The connectors are located on the inboard and outboard sides of the track where they are mounted on the protruding pins in the shoe assembly. The link is retained to the track shoe pins by a wedge, which fits into a cavity in the link and is bolted to the link. The wedge locks the pins rigidly to the link and keeps the pins from rotating.

The connector links are contoured to fit the shape of the sprocket teeth on the drive wheel. The drive torque is transferred to the track through contact between the connectors and the sprocket teeth. There are 160 connector links per track or 320 per vehicle.

2.6.1.2 Dimensional Requirements

The dimensional requirements for the end connector are specified on Drawing No. 11615320. Wedge Track Shoe, Drawing No. 8382359 and Bolt Drawing No. 8382360 are used for locking the connector to the track shoe pin. The connector is a steel forging of 4140 alloy or equivalent heat treated to Rockwell C 40-45. This is approximately equivalent to an ultimate tensile strength of 200,000 psi or 176,000 psi yield strength. The principal dimensions of the connector are shown in Figure 2-27.

2.6.1.3 Design Loads

The loads specified for the connector link is 11,700 lbs. maximum tension and 19,000 in. lbs. maximum torque on the track pin. The torque is caused by rotation of the track shoe relative to the pin as it travels around the drive wheel. The pin is mounted in the track shoe in a rubber sleeve which is bonded to both pin and shoe and it is the shearing of the rubber that produces the torque loads. This torque is transmitted to the connector link through the wedge.

It is difficult to correlate the tension loads on the connector with those on the sprocket tooth. The load specifications for the drive wheel sprocket, paragraph 2.3.1.3, indicate that the tension loads on one side of the track could be as high as 61,717 lbs. considerably higher than 11,700 lbs. Some of the tension load in the track can be taken by the track guide (Part No. 11599973) which is clamped to the track shoe pins in the center of the track. It seems unlikely, however, that the track guide is capable of withstanding the difference in these loads $(61{,}717 - 11{,}700) \cdot 2 = 100{,}000$ lbs.

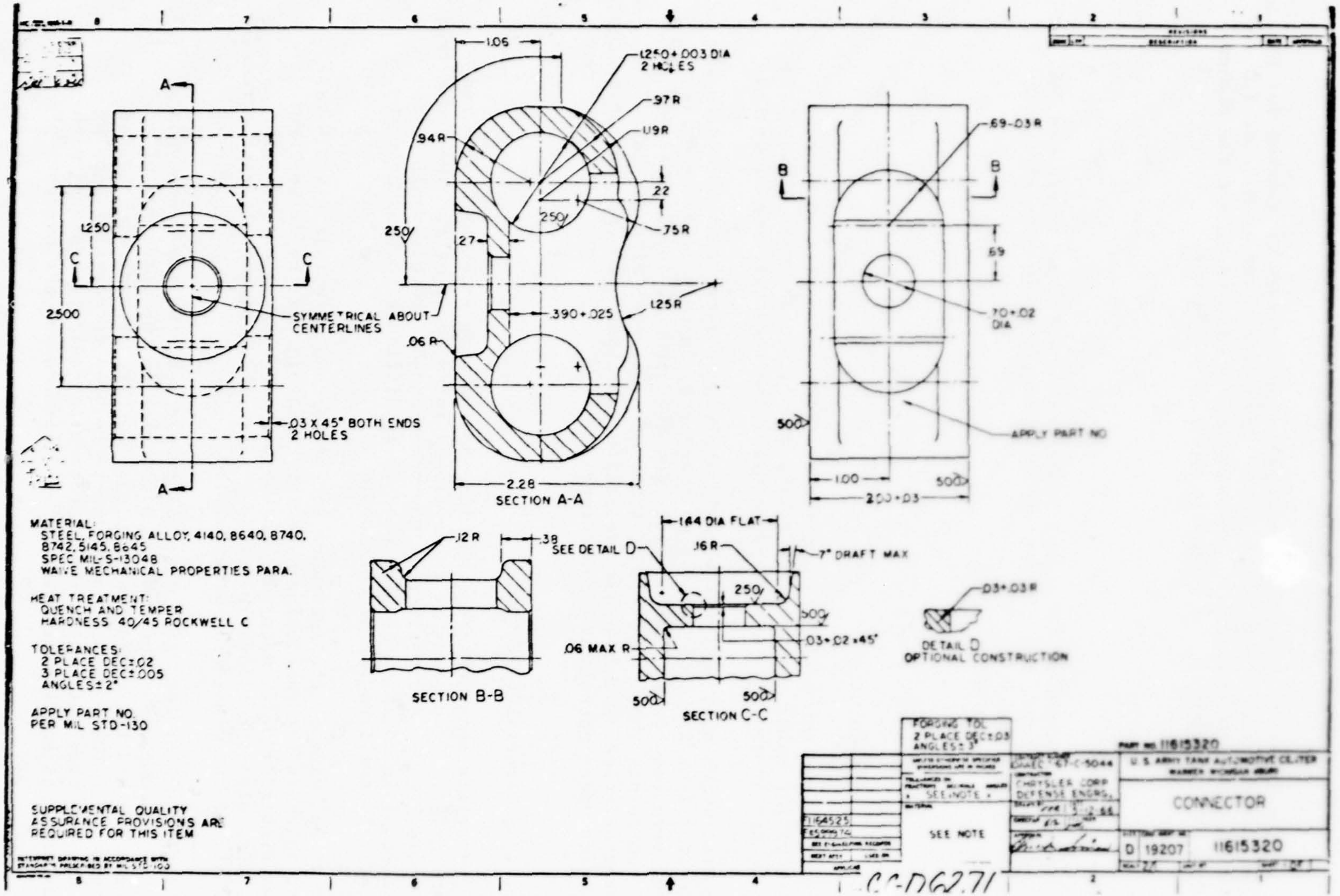


Figure 2-27 Connector Link Dimensions (Metal)

The stresses in the present connector are very low even if checked for the higher load of 61,717 lbs., f_t = 50,500 psi which gives an M.S. of 3.0 for ultimate tensile stress. It was therefore decided to use the higher load for the design of the composite material connector.

2.6.1.4 Weight

The specified weight of the steel connector is 2.6 lbs. There are 160 connectors per track and 320 per vehicle, resulting in a total weight of 832 lbs. per vehicle.

2.6.2. Feasibility Study

2.6.2.1 Introduction

The most efficient configuration for a fibrous composite component designed for mainly tension loads is a wound loop shown in Figure 2-28. The connector would be a suitable application for this type of design.

One area which must be considered in applying composite materials to the connector link is the effect on mating parts, in this case the drive wheel sprocket teeth. The analysis for the sprocket showed that one of the components should be made of steel or a material with similar characteristics. Other areas are stiffness compared to the present design and wear under operational conditions.

2.6.2.2 Material Selection

The materials considered for the loop design were graphite, Kevlar and glass fiber. Properties of these fibers in an epoxy matrix are summarized in Table 2-1. Boron fiber is not recommended for this application because of in addition to high cost it is difficult to bend over the relatively small radius of the loop.

For an estimate of the endurance stress for the materials R = .36 was used which is the same value used for the design of the drive wheel (See Figure 2-13). According to the Goodman diagram and assuming 50,000 cycles the allowable cyclic stress is 70% of the ultimate stress for all of the considered materials.

In reviewing the composite material properties, two groups of values should be compared. The tension allowables in the direction of the fibers (x) and the contact pressure allowable perpendicular to the fibers (y). The values should also be related to material density. It appears that the best overall properties could be achieved with graphite. It has the greatest "x" - stiffness and the highest "y" -

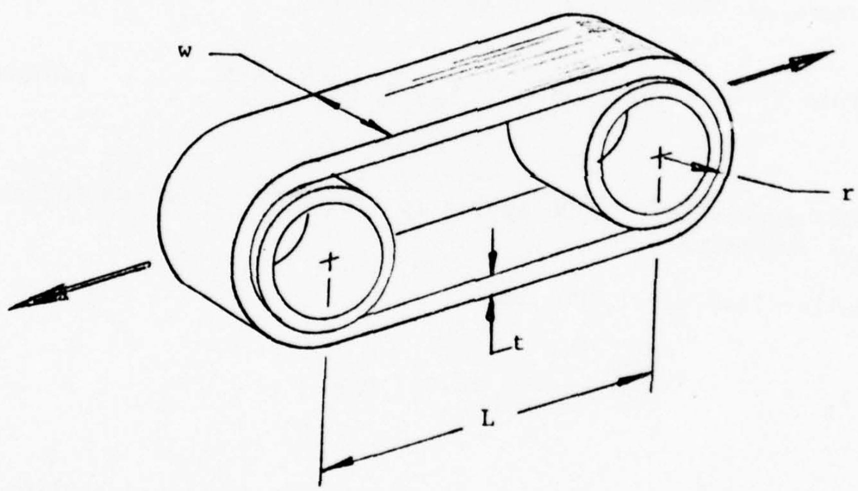


Figure 2-28 Fiber Tension Loop

proportional limit. In tension it is not as efficient as Kevlar, but Kevlar has a very low modulus perpendicular to the fiber.

2.6.2.3 Analysis

Previous research and experience has shown that all fibers in a loop in tension do not carry the same load. The load distribution to the individual fibers varies with the ratio t/r but in general the load capability of the loop is approximately 50% of the ultimate load of the straight fiber in tension.

Assuming that the external dimensions of the connector must remain the same as the present design and assuming that the present wedge configuration is maintained it can be seen that the continuous fibers must be divided into two loops, one on each side of the wedge. The available cross sectional area for each loop is: .31 x .31 in.

$$\text{For } P = 61{,}717 \text{ lbs.} \qquad f_t = \frac{61{,}717}{.31 \cdot .31 \cdot 4} = 160{,}000 \text{ psi}$$

The allowable stress for 50,000 cycles is $126{,}000 \cdot .5 = 63{,}000$ psi for graphite composite .

For the smaller load of 11,700 lbs,

$$f_t = \frac{11{,}700}{.31 \cdot .31 \cdot 4} = 30{,}437 \text{ psi}$$

which is well within the allowable stress, but as mentioned earlier it is believed that the low value of the load is incorrect.

The stiffness of the present design and the composite link is compared by comparing the parameter EA. The minimum cross section for the metal connector is 1.22 in^2.

$$EA_{stl} = 29 \cdot 10^6 \cdot 1.22 = 35.4 \cdot 10^6 \text{ lb. in.}^2$$

$$EA_{comp} = 21 \cdot 10^6 \cdot .31 \cdot .31 \cdot 4 = 8.1 \cdot 10^6 \text{ lb. in.}^2$$

or the stiffness of the composite material link is only 23% of the stiffness of the steel link. This means that a track with composite links would be more flexible under tension loads which may have a detrimental effect on its operation.

A check of the contact pressure between the link and sprocket teeth indicated that one of the surfaces must be made of steel. From a weight savings standpoint the biggest weight savings would be achieved by the use of composite links and steel sprockets but when one considers environmental conditions it appears that the connector link is exposed to a much more severe environment than the sprocket. Therefore it would be better if the sprocket was made of composite materials and the link remained steel.

Considering all factors it is concluded that it is not feasible to replace the existing link with one made of fibrous composite materials.

2.7 Truck Wheel Assembly

2.7.1 Baseline Requirements

2.7.1.1 Description

The truck wheel is designed for a five ton, 8 x 8, cargo truck. There are eight wheels, plus a spare per truck. The wheel is designed for a tube type Michelin 16 - 20x5 tire with a nominal tire pressure of 45 psi.

2.7.1.2 Dimensional Requirements

The dimensions of the truck wheel components are specified on Drawings:

DTA	-	169825	Wheel Assembly (Tube Type)
DTA	-	10945214	Hub, Wheel
DTA	-	10945234	Stud, Hub and Drum

Drawing DTA-11601146 - "Brake Drum and Drive Flange Assy" is supplied for reference only. This part is not included in the study. Sixteen studs 10945234 and used to attach the brake drum flange to the Hub 10945214. The Wheel Assembly 169825 is attached to the hub by means of eight unspecified fasteners.

Two components, wheel assembly (169825) and hub (10945214), are to be designed from composite materials, such, that they can be assembled together with the brake drum and drive flange assembly,11601146.

The current wheel assembly, 169825, is welded and consists of six parts:

1. Spacer, DTA-169825-1, .31 thick made from 1010-1012 hot rolled steel (welded).
2. Base, DTA-169825-2, made from Goodyear Part No. B1020 M RIM (welded).
3. Flange, DTA-169825-3, made from Goodyear Part No. B1020 M RIM (welded).
4. Disc, DTA-169825-4, made from 1010-1012 hot rolled steel (welded).
5. Flange, Goodyear Part No. F 1020 M.
6. Lock Ring, Goodyear Part No. LR 20 M

The above Goodyear drawings were not provided.

The current wheel hub, 10945214, is an aluminum forging, 6151-T6 per QQ-A-367. The drawing shows only the machined dimensions, but not the basic forging dimensions.

The specified design objectives are:

Increased Impact Resistance

Light Weight

2.7.1.3 Design Load Conditions

The only specified load requirements are:

Maximum static vertical loads, 4,500 lbs. which is based on vehicle weight of 36,000 lbs. uniformly distributed on all eight wheels. No maximum dynamic vertical load or side load is provided. It is stated that the composite material strength should be equal to 1010-1012 hot rolled steel.

Fatigue cycle requirements, maximum tire pressures and braking loads are not provided.

Due to lack of above load information, an estimate of design loads was made. The basis for the estimate was the estimated ultimate strength of the metal components of the wheel. Some dimensions are given on the drawings, other dimensions were scaled from the available reduced size drawings.

The material properties of the hot rolled 1010-1012 type steel used for the wheel hub are (Ref: Ryerson Data Book).

$$F^{tu} = 47{,}000 \text{ psi}$$

$$F^{ty} = 30{,}000 \text{ psi}$$

$$F^{su} = 30{,}000 \text{ psi}$$

$$E = 29 \cdot 10^6 \text{ psi}$$

$$G = 11 \cdot 10^6 \text{ psi}$$

$$\gamma = .283 \text{ lb/in}^3$$

The approach to determine ultimate tire pressure loads on the wheel is to calculate stresses with a unit load and then to modify the unit load by a factor which would result in ultimate material stresses in the steel wheel. Since the same method of analysis will be used for the composite wheel, it is expected that the designs will be equivalent.

1. Tire Pressure Loads - According to information received from a Michelin tire dealer, the standard pressure of the Michelin 16-20 tire is 45 psi. A unit load of 100 psi will be used for analysis of the steel wheel. The assumed tire dimensions are shown in Figure 2-29.

The tire pressure is loading the wheel rim in compression and the rim flange in bending and shear.

Rim Compression:

$$f^c = \frac{p \cdot R}{2 \cdot t}$$

$$= \frac{100 \cdot 10}{2 \cdot .31} = 1{,}613 \text{ psi}$$

For a material yield stress of 30,000 psi, the compressive failure would start in the wheel rim at 30,000 · 100/1,613 = 1,860 psi tire pressure.

2. Rim Side Loads - The tire pressure is resisted laterally by the rim and by the tire crown. The projected area of the torus (inside) is:

$$\frac{\pi}{4}\left(46^2 - 20^2\right) = 1{,}348 \text{ in.}^2$$

At 100 psi tire pressure, a total side load of 134,800 lbs. is generated, half of which is reacted by the rim. The pressure centroid is at the mean radius of the flange (10.8 in.) and the pressure is uniformly distributed over the flange height. The load per inch of circumference at 100 psi tire pressure:

$$W = \frac{134{,}800}{2 \cdot 2\pi \cdot 10.8} = 933 \text{ lb/in}$$

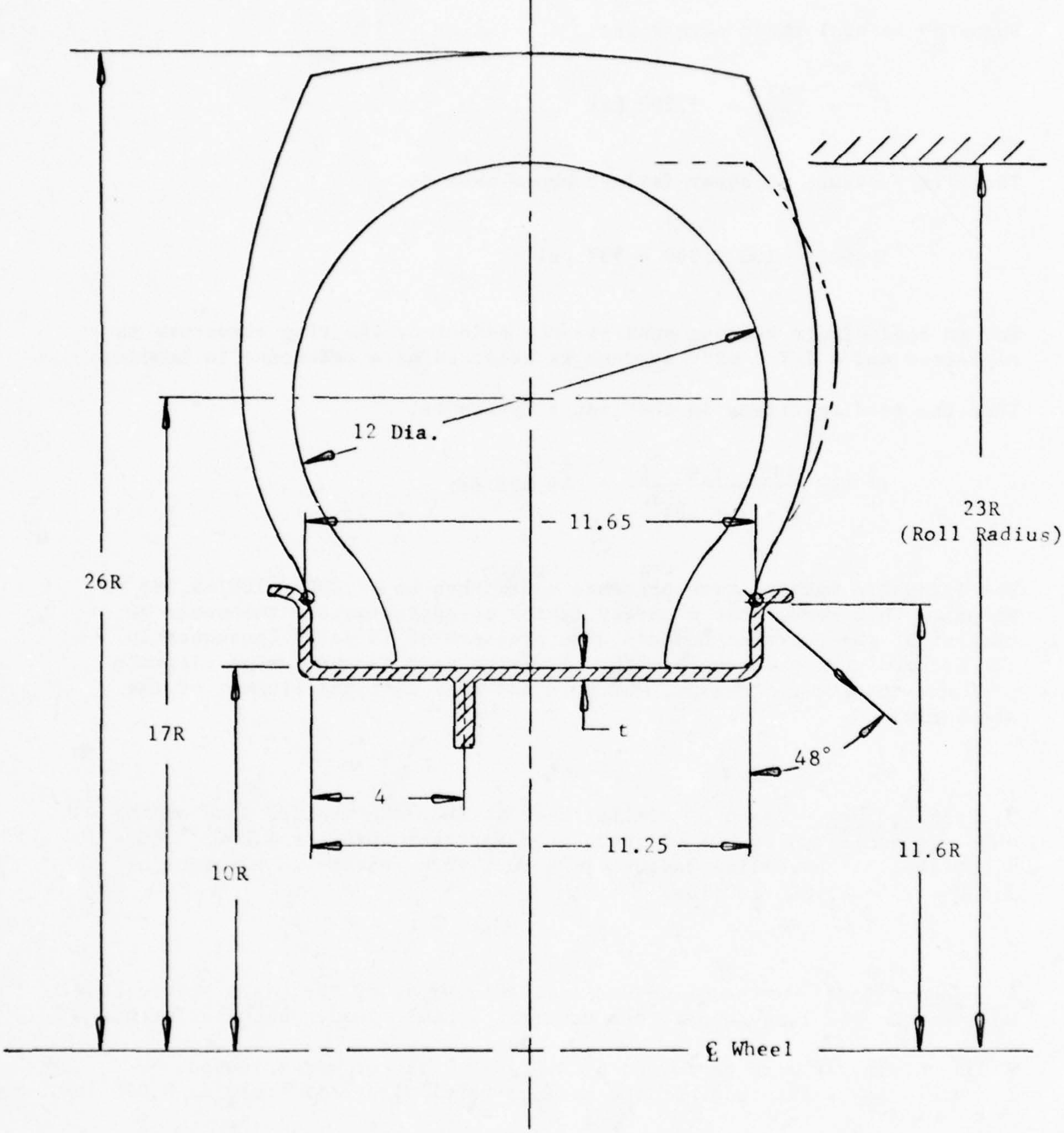


Figure 2-29 Truck Wheel Dimensions

Then the lateral shear stress is:

$$f^s = \frac{993}{.31} = 3,202 \text{ psi}$$

The tire pressure at shear failure would then be,

$$30,000 \cdot 100/3,202 = 937 \text{ psi}$$

For an approximate bending analysis the effect of the ring structure is neglected and a 1 in. wide segment is selected as a reference in bending.

Then the bending stress in the flange corner is,

$$f^b = \frac{993 \cdot 1.6 \cdot 6}{2 \cdot 1 \cdot .31^2} = 49,598 \text{ psi}$$

The allowable maximum tire pressure would then be 47,000 · 100/49,598 = 95 psi. This means that a safety factor of approximately two would be present at the reported nominal tire pressure of 45 psi. Consequently the assumed unit pressure of 100 psi can be used as the design ultimate pressure for the rim design, which is the most critical element of the steel rim.

3. Braking Load - Based on similar work at REC, the braking load on the wheel is estimated to 80% of the static vertical load, or 4,500 · .80 = 3,600 lbs. At a rolling radius of 23 in., this results in a torque of 3,600 · 23 = 83,000 in. lbs.

4. Dynamic Loads - The given vertical load of 4,500 lbs. is a static load. However the wheel may experience vertical dynamic loads which are greater.

Based on similar work performed by REC, a 2 g factor was selected. Therefore it is assumed that the maximum vertical dynamic load is 9,000 lbs.

For the side load a factor of 50% is usually applied. Therefore the maximum side load is 2,250 lbs. It could occur simultaneously with the 4,500 lbs. static load and tire pressure.

5. <u>Fatigue Loads</u> - For an estimate of material fatigue allowables it is assumed, that the life expectancy of a wheel is 100,000 miles at an average speed of 30 mph. The load cycle is completed after one revolution of the wheel, during which the loads on the rim changes from maximum vertical load to zero load with the tire pressure load remaining approximately constant. At a rolling diameter of 46 in., the wheel covers, during one revolution, $\pi \cdot 46 = 144.5$ in. = 12.04 ft. At a speed of 30 mph the wheel rotates $30 \cdot 5280/12.04 = 13,156$ times/hour. Therefore the load change frequency is $13,156/3,600 = 3.65$/sec. During the life of 100,000 miles the number of load cycles is $100,000 \cdot 5,280/12.04 = 44 \cdot 10^6$ cycles, meaning that the wheel should be designed to the endurance limit of the material, which is usually reached at $2 \cdot 10^6$ cycles.

Based on the above considerations and estimates, the following load requirements were used by REC for the design of the composite wheel:

Static Vertical Load	4,500 lbs. limit load
Dynamic Vertical Load	9,000 lbs. limit load
Side Load	2,250 lbs. limit load
Tire Pressure	50 psi, limit load
Brake Moment	82,000 in. lbs. limit load
Fatigue Vertical Load	0 to 4,500 lbs.
Endurance Fatigue Cycling	$2 \cdot 10^6$ psi

Limit loads should have a safety factor of a least "two" for ultimate material strength. Fatigue loads should have a safety factor of at least "one" for material endurance limit. The tire pressure loads are superimposed on all load conditions.

2.7.1.4 Weight of the Current Wheel

No information is given on the weight of the current metal or of its components.

Therefore an estimate of the current wheel weight was prepared.

Wheel Assembly (169825)

Basic Rim	=	61.0 lbs.
Fixed Flange	=	16.2 lbs.
Removable Ring	=	37.5 lbs.
Hub Mounting Flange	=	8.9 lbs.
Total:		123.6 lbs.

Wheel Hub (10945214)	20.9 lbs.

2.7.2 Feasibility Study

2.7.2.1 Introduction

The wheel and hub assemblies were designed for four different load conditions.

1. Vertical Load 4,500 lbs.
   Side Load 2,250 lbs.
   Tire Pressure 50 psi

2. Vertical Load 9,000 lbs. (Ult.)
   Tire Pressure 50 psi

3. Vertical Load 4,500 lbs.
   Tire Pressure 50 psi
   Cyclic Load

4. Tire Pressure 100 psi (Ult.)

The tire pressure loads are imposed on the wheel rim only. The loads are limit loads except as noted.

Several concepts were investigated. The most feasible design is a rim assembly made from continuous graphite fiber/epoxy composite and a hub made from compression molded short glass fiber sheet molding compound.

2.7.2.2 Material Selection

The material selected for the wheel assembly is AS-graphite/epoxy tape and fabric. The properties are shown in Table 2-12. In the text the fiber orientations are designated:

L = hoop (circ.)

M = axial (radial in flanges)

N = $\pm 45^\circ$ to wheel axis

For the hub design sheet molding compounds S-6414 by Fiberite Corporation and XD-901_ manufactured by Dow Chemical Company were considered. The material properties used in the analysis are shown in Table 2-13. The resin system in these materials is polyester. The strength is somewhat lower than for epoxy systems but advantages are lower cost and lower manufacturing cost due to lower molding pressures. Sheet molding compounds also have smaller bulk factors than bulk molding compounds which means that the molds can be made simpler. Sheet molding compounds have successfully been used in the automotive industry for a number of years.

2.7.2.3 <u>Analysis of Wheel Assembly on Rim Flange</u>

Load Condition 1	Vertical Load	4,500 lbs.
	Side Load	2,250 lbs.
	Tire Pressure	50 psi

The vertical and side loads are assumed reacted at the contact radius of 11.6 in. (See Figure 2-29) and the pressure load at a mean radius of 10.8 in.

It is also assumed that the vertical loads from ground contact are concentrated between $+45^\circ$ and -45° from the vertical centerline or equivalent to an arc length of 18.2 in. at 11.6 in. radius. The centroid of this arc is located 10.47 in. from the wheel centerline.

Vertical Load

$$P_V = \frac{4,500}{2} + \frac{2,250 \cdot (23 - 10.47)}{11.65} = 4,670 \text{ lbs.}$$

reacted at 11,6 radius.

Table 2-12

Material Properties for AS Graphite Tape and Fabric in an Epoxy Matrix

		0° Tape	±45° Fabric	0°/90° Fabric
F^{tu}	PSI	180,000	27,300	80,000
F^{cu}	PSI	180,000	30,000	88,000
F^{su}	PSI	12,000	48,000	19,000
E^{t}	PSI	$21 \cdot 10^6$	$3.2 \cdot 10^6$	$10.3 \cdot 10^6$
E^{c}	PSI	$21 \cdot 10^6$	$2.8 \cdot 10^6$	$8.5 \cdot 10^6$
G	PSI	$.65 \cdot 10^6$	$4.5 \cdot 10^6$	$1.04 \cdot 10^6$
γ	lb/in^3	.056	.057	.057
ϵ^{tu}	in/in	.0086	.0085	.0078
ϵ^{cu}	in/in	.0086	.0107	.0103
δ	rad/rad	.0185	.0107	.0183
$t_{ply, cured}$	in.	.0055	.010	.010

Table 2-13

Material Properties for Sheet Molding Compounds

F^{tu} psi	25,000
F^{cu} psi	30,000
F^{b} psi	35,000
E^{b} psi	$2.5 \cdot 10^{6}$
F^{ils} psi	5,000
F^{ips} psi	7,500
F^{br} psi	20,000
γ lb/in^{3}	.069

Due to the flange contour, the vertical load produces a side component load of:

$$P_{S_1} = \frac{P_V}{tg\ 48^\circ} = 4,205 \text{ lbs.}$$

Side loads on flange,

$$P_{S_2} = 2,250 \text{ lbs.}$$

Pressure load on flange at 50 psi acting at a mean radius of 10.8 in. is,

$$P_{S_3} = \frac{1,348 \cdot 50}{4 \cdot \pi \cdot 10.8} = 497 \text{ lbs/in}$$

These side loads cause a bending moment on the flange. For a one inch arc the total bending moment at the neutral axis of the .465 in. thick flange corner is:

$$M_{b_{max}} = 1,134 \text{ lbs/in}$$

The total side load on the flange is 852 lbs/in for Load Condition 1

Load Condition 2	Vertical Load	9,000 lbs.
	Tire Pressure	50 psi

$$P_V = 4,500 \text{ lbs.}$$

$$P_{S_1} = 4,052 \text{ lbs.}$$

$$P_{S_3} = 497 \text{ lbs/in}$$

Bending Moment

$$M_{b_{max}} = 892 \text{ in. lbs/in}$$

The total side load on the flange = 720 lbs/in for Load Condition 2

Load Condition 3 Vertical Load 4,500 lbs.

Tire Pressure 50 psi

Cycling

$$M_{b_{min}} = 456 \text{ in. lbs/in}$$

$$M_{b_{max}} = 674 \text{ in. lbs/in}$$

$$\text{Then } R = \frac{456}{674} = .68 \text{ (cycle load factor)}$$

Side Load P = 497 to 608 lb/in

Load Condition 4 Tire Pressure 100 psi (ult.)

Side Load = 2 · 497 = 994 lbs/in (on rim)

2. Analysis of the Flange - A concept of the wheel assembly is shown in Figure 2-30. The basic laminate of the rim is .31 in. thick and consists of 40 interspersed plies.

20 plies longitudinal tape $t_1 = 20 \cdot .0055 = .11$ in.

20 plies 45° tape $t_2 = 20 \cdot .0100 = .20$ in.

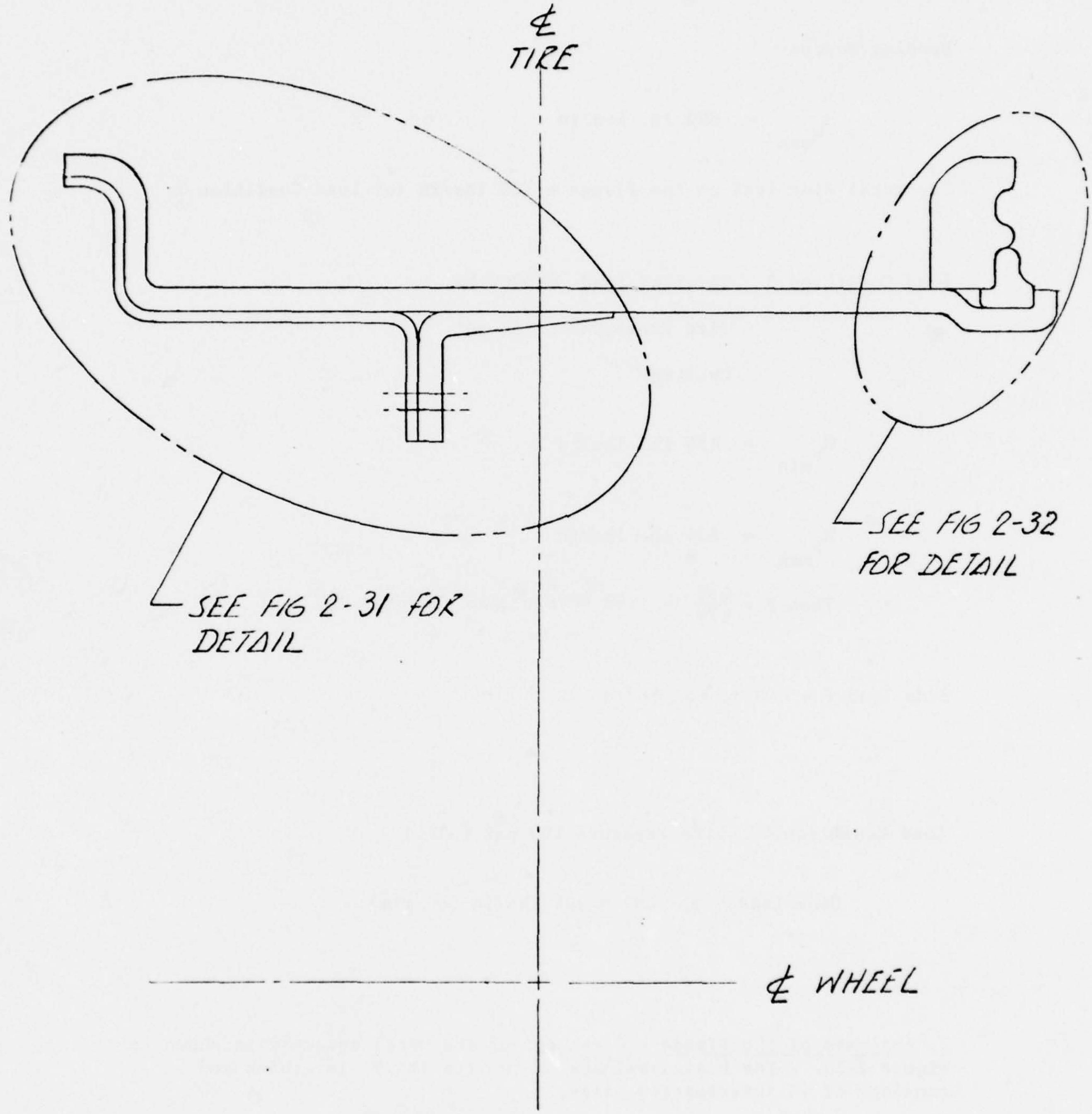


Figure 2-30 Composite Wheel Design Concept

To determine the effective moment of inertia of a one inch wide strip, the ±45° fabric layers were converted to 0° tape layers by reducing the effective width the ratio of material moduli, or 3/21 = .143.

The moment of inertia of this 0° tape section was calculated to,

$$I_1 = .001037 \text{ in}^4$$

To compensate for stresses due to the lateral bending moment, the rim thickness was increased by an additional 10 plies of tape and 10 plies of fabric, resulting in a total nominal thickness of the laminate of .465 in. The additional plies were added between the solid flange and the inner mounting flange (See Figure 2-31).

The moment of inertia of this section converted to 0° tape was calculated to,

$$I_2 = .003494 \text{ in.}^4$$

The outer flange consists of the .465 in. thick laminate. For the stress analysis the S-shaped flange includes all materials outside of the point of tangency between the cylindrical section and the bend radius (See Figure 2-31). The moment of inertia of the S-shaped flange was calculated with respect to its centroid,

$$I_3 = .6345 \text{ in.}^4$$

3. <u>Bending of Flange</u> - Load Condition 1 produces the maximum bending moment on the flange 1,371 in. lbs/in. and a lateral shear force of 852 lbs/in.

The analysis is carried out per Timoshenko, "Strength of Materials". Part Part II, 3rd Edition, p. 142.

$$\beta = \sqrt[4]{\frac{3\ (1 - \mu^2)}{c^2 \cdot h_1^2}}$$

Where c = inner radius of the rim = 9.696 in.

$\beta = .61$

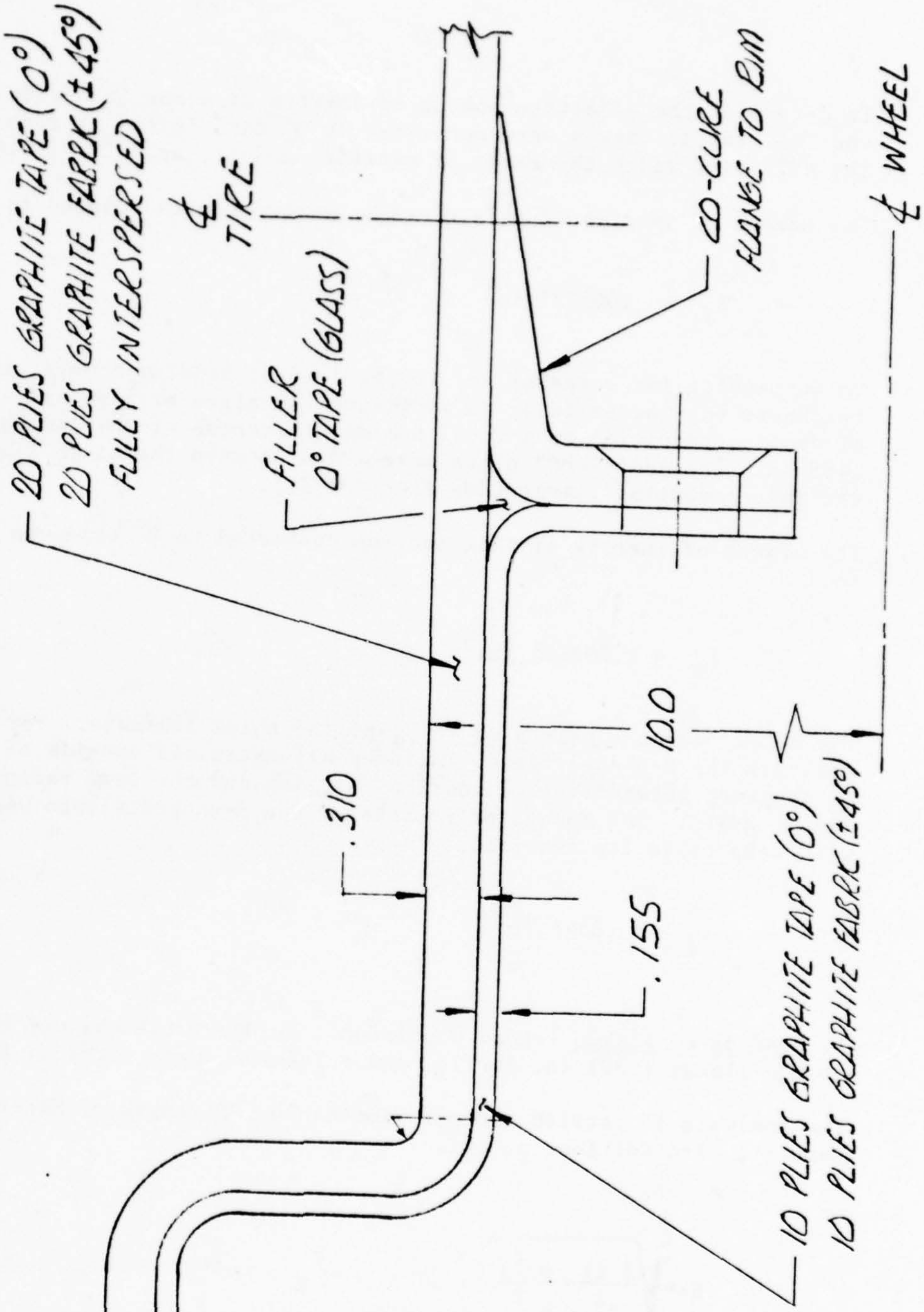


Figure 2-31 Detail of Rim - Fixed Flange

and

$$\ln \frac{d}{c} = .1442$$

where $d = 11.20$ in.

and $\beta \frac{h}{2} = .1418$

Then bending moment at the flange/cylinder intersection,

$$M_o = P\,(d-c)/K$$

where P = lateral load = 852 lbs.

$$K = 1 + \beta \frac{h}{2} + \frac{1 - u^2}{2\beta c} \left(\frac{h}{h_1}\right)^3 \ln \frac{d}{c}$$

$$K = 1.15$$

and the maximum bending moment for Load Condition 1,

$$M_o = 852 \cdot 1.504/1.15 = 1{,}113 \text{ in. lbs/in}$$

and the maximum bending stress in the all 0° tape section:

$$f^b = \frac{M_o \cdot t}{I_2 \cdot 2}$$

$$f^b = 74{,}062 \text{ psi}$$

Which gives a safety factor of 180,000/74,062 = 2.4 for ultimate strength of a 0° tape laminate.

At this stress the maximum strain at the surface of the laminate is,

$$= \frac{74,062}{21 \cdot 10^6} = .0035 \text{ in/in}$$

For the actual laminate, the allowable stress is referenced to a load in the direction of the longitudinal tape and 45° fabric. For a fiber orientation of 0°/45° at a ratio of 35/65% the allowable material strength is (per Air Force Design Guide Vol. 1),

$$F^{cu} = 70,000 \text{ psi}$$

$$F^{tu} = 73,000 \text{ psi}$$

At a laminate hybrid modulus of $7 \cdot 10^6$ psi the applied stresses at .0035 in/in strain are:

$$f^b = .004 \cdot 7 \cdot 10^6 = 24,500 \text{ psi}$$

Considering also the axial load of 852 lbs., an additional tensile stress of 852/.465 = 1,832 psi is applied, resulting in a total tensile stress of 26,300 psi. Then the safety factor against tensile failure is:

$$SF = \frac{73,000}{26,300} = 2.78$$

The shear force across the rim cylinder at the flange joint is:

$$P_o = \beta \cdot M_o$$

$$P_o = 680 \text{ lbs/in}$$

For a rim thickness of .465 in., the shear stresses are then:

$$f^s = \frac{680}{.465} = 1,460 \text{ psi}$$

and the interlaminar shear allowable stress is at least 8,000 psi.

Therefore the bending and shear stresses in the flange are within the allowable stresses and have a safety factor against material ultimate of two or more. In order to achieve the material strength values, it is important that the laminating methods are such that a dense and void free laminate is achieved, particularly in the transition radius between flange and cylinder.

For Load Condition 3 the estimated fatigue allowables are,

> For a 180,000 psi ultimate stress of unidirectional graphite, the allowable stress at $2 \cdot 10^6$ (endurance) is 46% of ultimate at R = 0.1, or 82,800 psi. For R = .68 the allowable is 74% of ultimate or 133,200 psi.

The applied maximum moment for Load Condition 3 is 674 in. lbs/in. Since the applied maximum moment in Condition 1 is 1,134 in. lbs/in and modified, more accurately, to 1,113 in. lbs/in, the moment for calculating bending stresses for Condition 3 is,

$$M_o = \frac{1,113}{1,134} \cdot 674 = 661 \text{ in. lbs/in}$$

This results in a stress of the modified 0° tape section of,

$$f^b = \frac{661 \cdot .465}{.003494 \cdot 2} = 43,985 \text{ psi}$$

at the strain of $43,985/21 \cdot 10^6 = .002$ in/in which is only about 25% of ultimate strain. For R = .68 74% of ultimate strain is acceptable.

By inspection it is appearant that the bending loads imposed on the flange by Load Condition 4 are less than for Load Condition 1 and is therefore not critical.

4. Flange in Plane Bending - Calculation of bending coefficients K is done in the same manner as for the support roller in Paragraph 2.4.2.3. The vertical load is represented by two concentrated loads, each assumed to be 23° from the symmetry line. Therefore $\theta = 23°$.

Then,

$$\theta = .4014 \text{ rad}$$

$$\sin\theta = .3907$$

$$\cos\theta = .9205$$

$$\cos^2\theta = .8473$$

Maximum bending moment at $\theta = \alpha = 23°$

$$M_{max} = Q \cdot R \cdot K_1$$

$$K_1 = .23868 \cdot \cos\alpha - .50 \sin\theta + .15915 \cdot K_2$$

$$= .0966$$

$$K_2 = \alpha \cdot \sin\alpha + \theta \cdot \sin\theta + \cos\theta - \cos\alpha \cos^2\theta$$

$$= .4542$$

It is assumed that the bending load is resisted at the flange alone. The moment of inertia for the flange, $I_3 = .6346$ in.^{4}.

The maximum vertical load on the flange is for Condition 2, where $P_V = 4{,}500$ lbs. Then the bending moment at the centroid of the flange is at 23° from the vertical centerline:

$$M = \frac{P_V}{2} \cdot R \cdot K_1$$

$$= 2{,}365 \text{ in. lbs.}$$

and the maximum bending stress at the outer diameter of the flange,

$$f = \frac{2,365}{.6346} = 3,727 \text{ psi}$$

The safety factor for in-plane bending for Condition 2 is then,

$$S.F. = \frac{22,289}{3,727} = 6$$

5. Analysis of Removable Ring Flange - Load Condition 1 was determined to be the most critical for the lateral loading of the flange.

The maximum lateral load per in. arc length is therefore:

Side load due to road loads,

$$P_1 = \frac{4,205 + 2,250}{18.2} = 355 \text{ lb/in}$$

reacted at R = 11.6 in.

Side load due to pressure load, 50 psi,

$$P_3 = \frac{33,700}{2\,\pi \cdot 10.8} = 497 \text{ lb/in}$$

reacted at R = 10.8 in.

These loads are resisted by the lock ring at a radius of approximately 10.25 in.

The resultant of these loads is at R = 11.13. The twisting moment is then,

$$M_T = 750 \text{ in. lbs/in}$$

and the lateral shear force is 852 lbs/in. which is loading the lock ring in shear and the removable flange in bending and shear.

The above calculations are for a ring with rectangular cross section. The actual flange has a more complex shape as shown in Figure 2- 32 Approximate methods indicate that the stresses in the flange, consisting of circumferential fibers as well as fabric oriented at $\pm 45°$, are similar to the rectangular cross section ring calculated above.

6. Stresses in Wheel Rim Cylinder - The vertical shear transmitted through the rim cylinder is for Load Conditon 1,

$$P_V = \frac{4,500}{2} + \frac{2,250 \cdot 11.4}{11.65} = 4,452 \text{ lbs/in}$$

The maximum shear flow at $\alpha = 90°$ is,

$$S = \frac{P_V}{\pi \cdot R_m} \sin \alpha$$

$$= 144 \text{ lb/in}$$

The shear stress in the 45° plies is,

$$f^s = \frac{144}{.20} = 720 \text{ psi}$$

Since F^s = 48,000 psi, the safety factor for a shear failure is large.

For Load Condition 2 a braking moment of 83,000 in. lbs. is combined with the dynamic vertical load of 9,000 lbs. These loads are introduced at both flanges of the cylinder. The maximum shear flow,

$$S_{max} = 214 \text{ lb/in}$$

and the shear stress in the 45° plies,

$$f^s = \frac{214}{.20} = 1,068 \text{ psi}$$

also for this condition the safety factor is high.

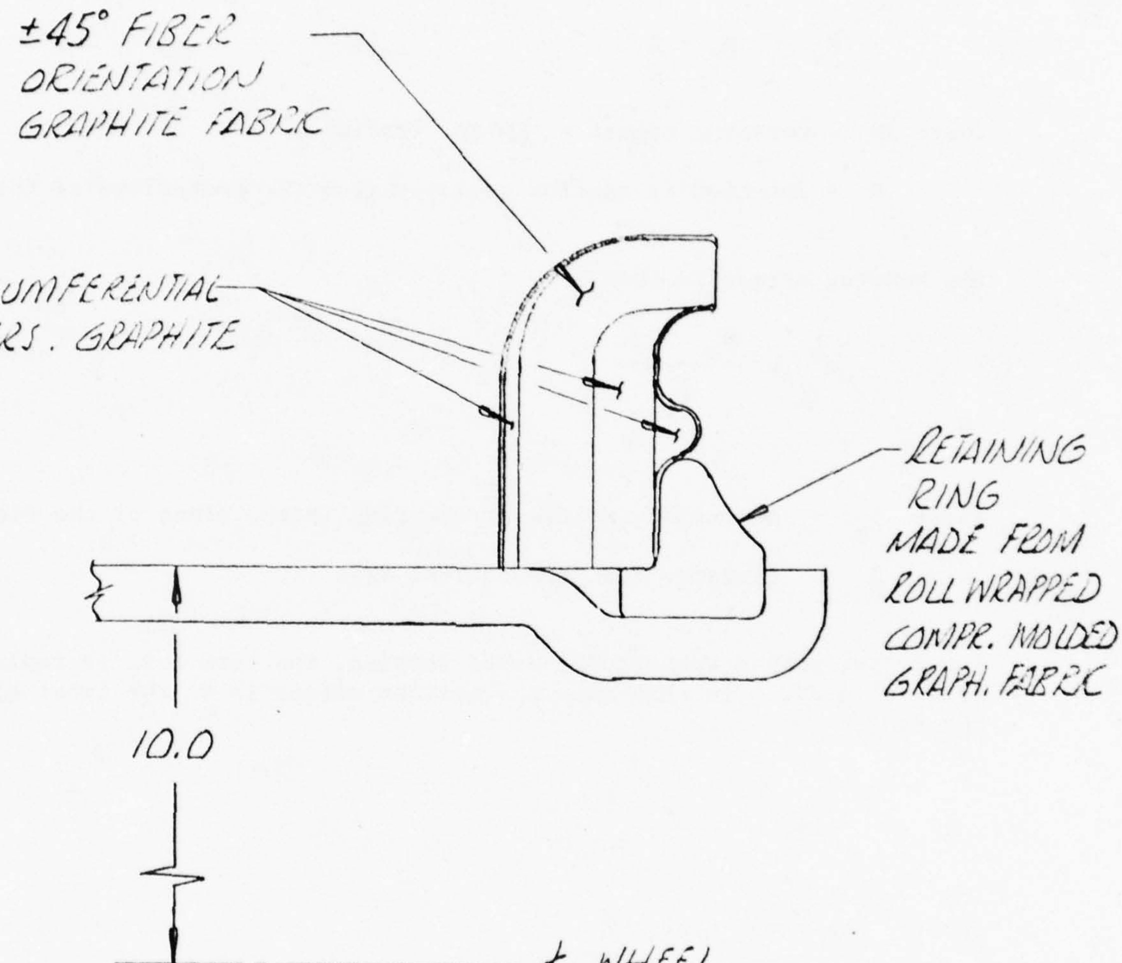


Figure 2-32 Detail of Rim - Removable Ring and Lock Ring

Per Timoshenko, "Strength and Materials", Part II, 3rd Edition, p. 138, the bending moment at the cross section of the flange is,

$$M_o = M_T \cdot R$$

where M_T = twisting moment = 750 in. lbs/in

R = location of section centroid from the centerline of the wheel

The bending stress is then,

$$f^b = \frac{M_o \cdot X}{I_y}$$

Where I_y = moment of inertia of the ring in the plane of the ring

X = distance from the neutral axis

For a ring with a rectangular cross section, the term X/I_y is replaced by $12/h^3$ r ln d/c. In this case the maximum stress is at the inner corner of the ring,

$$f^b_{max} = \frac{6 \cdot M_t \cdot R}{h^2 \cdot c \cdot \ln \frac{d}{c}}$$

where h = ring thickness

c = ring inside radius

d = ring outside radius

Assuming c = 10 in. and d = 11.89 in. R = 10.945 in.

then h_{min}(required) = .56 in., make .60 in.

and

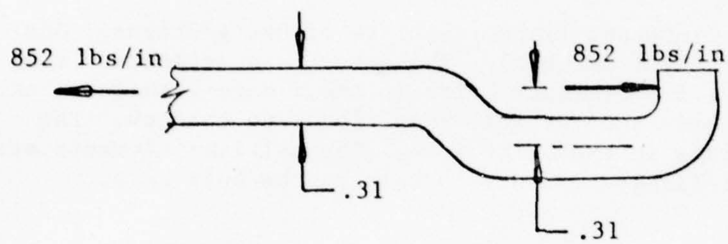
$$f^b = 75{,}823 \text{ psi}$$

Compression of the rim cylinder is caused by tire pressure, 100 psi ultimate. The compressive stress is,

$$f^c = \frac{100 \cdot 9.845}{.31} = 3,176 \text{ psi}$$

The allowable stress in the hoop direction is 22,000 psi.

The loads on the lockring ring recess are shown below.



The maximum bending stress for a one inch wide section,

$$f^b = \frac{852}{.31} + \frac{852 \cdot .31 \cdot 6}{.31^2}$$

$$= 19,239 \text{ psi (tension)}$$

The modulus of elasticity for the hybrid laminate consisting of 35% 0° and 65% ±45° fibers = $9.3 \cdot 10^6$ psi. The maximum strain is .0021 in/in. This corresponds to 43,400 psi allowable stress in the longitudinal tape and 6200 psi in the ±45° fabric. The safety factors are therefore large.

The shear stress,

$$f^s = \frac{852}{.31} = 2748 \text{ psi}$$

The allowable shear stresses are approximately 8,000 psi.

The tension stresses in the rim are caused by tire pressure and the side component of the vertical load. The lateral load was previously determined to be 852 lb/in (limit).

Assuming that mainly the longitudinal tape is resisting this load, the stresses are,

$$f^t = \frac{852}{.11} = 7,745 \text{ psi}$$

The safety factor is very large since the allowable tension stress is 180,000 psi.

7. Internal Flange - The internal flange consists of two sections. One is formed by a part of the rim material. The other is a separate ring bonded to the rim. Eight 5/8 diameter bolts in the flange transfer shear forces caused by brake loads and vertical wheel loads to the hub. The flange has cutouts for lugs in the brake drum. The critical elements are therefore the individual flange segments containing the bolt holes.

The maximum shear flow at the bolt radius of 8.562 in. is produced by,

1. Maximum vertical load 9,000 lbs.
2. Static vertical load 4,500 lbs. with brake moment 83,000 in. lbs.

The lateral shear is produced by 2,250 lbs. side load in combination with the 4,500 lbs. vertical load.

The maximum shear flow for Condition 1 is,

$$S = \frac{9,000}{\pi \cdot 8.562} = 335 \text{ lb/in}$$

and for Condition 2,

$$S = \frac{4,500}{\pi \cdot 8.562} + \frac{83,000}{8.562^2 \cdot 2\pi}$$

$$= 348 \text{ lb/in}$$

The load per bolt is,

$$P^S = 348 \cdot 6.72 = 2,335 \text{ lbs.}$$

Since the flange transfers mainly shear, the fiber orientation should include $\pm 45°$ tangential to the bolt circle.

As shown on Figure 2-31 the internal flange is built up of ten plies radial tape (.055 in.) and ten plies $\pm 45°$ fabric (.100 in.) plus the .345 thick bonded on flange which consists of $\pm 45°$ fabric.

The bearing pressure is calculated from:

$$P_{BR_{max}} = .583 \sqrt{\frac{P}{L} \left(\frac{1}{R_2} - \frac{1}{R_1}\right) \Big/ \left(\frac{1}{E_1} + \frac{1}{E_2}\right)}$$

where

P = bearing load = 2,335 lbs.

L = bearing length = .445 in. (only $\pm 45°$ fibers)

R_1 = radius of the hole = .35 in.

R_2 = radius of the bolt = .313 in.

E_1 = modulus of composite = $2.8 \cdot 10^6$ psi

E_2 = modulus of bolt = $29 \cdot 10^6$ psi

Then

$$P_{BR_{max}} = 39,105 \text{ psi}$$

which is higher than the allowable F^{cu} = 30,000 psi. The bearing stress may be reduced by adding metal shims to the laminate or add metal insert to the holes. The latter approach was chosen and steel bushings with .85 in. O.D. and .70 in. I.D. were added to the holes.

2.7.2.4 Design Analysis of the Hub (10945214)

A concept of the hub is shown in Figures 2-33 and 2-34. The structural function of the hub is to transmit road loads from the rim to the axle, and the drive and brake torques from the brake drum (11601146) into the wheel rim. The torque affects only the portion of the hub, which is between the bolt patterns for the wheel rim (8.562 dia.) and the brake drum (8.25 dia.). The center of the hub transmits only vertical and side loads. The wheel is attached to the hub by eight 5/8 dia. bolts. The brake drum is held to the hub by sixteen 7/16 in. dia. studs. Because of close tolerance requirements for the bearings, a metal insert has been added to the center of the hub. It is molded in place.

The present hub is an aluminum forging 6151-T6 with the following properties (ANC-5, March 1955).

F^{tu}	44,000 psi
F^{ty}	37,000 psi
F^{su}	28,000 psi
E	$10.2 \cdot 10^6$ psi
G	$3.85 \cdot 10^6$ psi
γ	.097 lb/in^3

1. Stress Analysis - The hub is molded from sheet molding compounds such as S-6414 or XD-3013. Properties are listed in Table 2-13 in Paragraph 2.7.2.2. The side load, 2,250 lbs, is introduced at a rolling radius of 23 in. and is distributed over a section of $\pm 30°$ form the vertical centerline.

$$\Delta P_L = 706 \text{ lb/in at the bolt circle}$$

Roark, "Formulas for Stress and Strain", 4th Edition, p. 222, Case 20 was used for the analysis of the hub.

B = .61

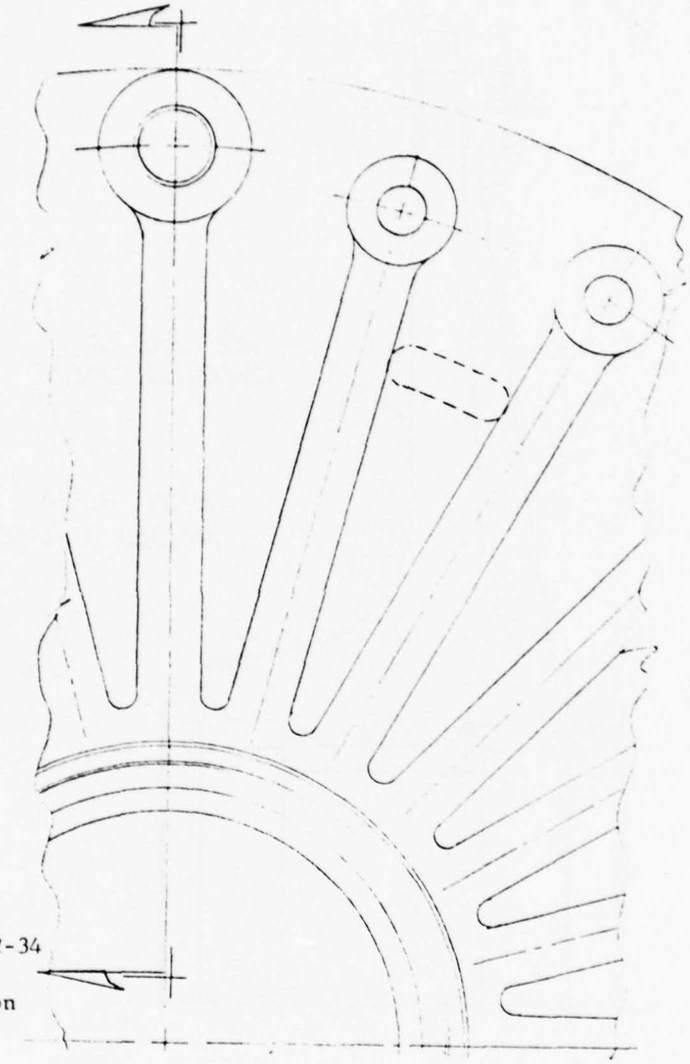


Figure 2-33 Composite Wheel Hub Design Concept

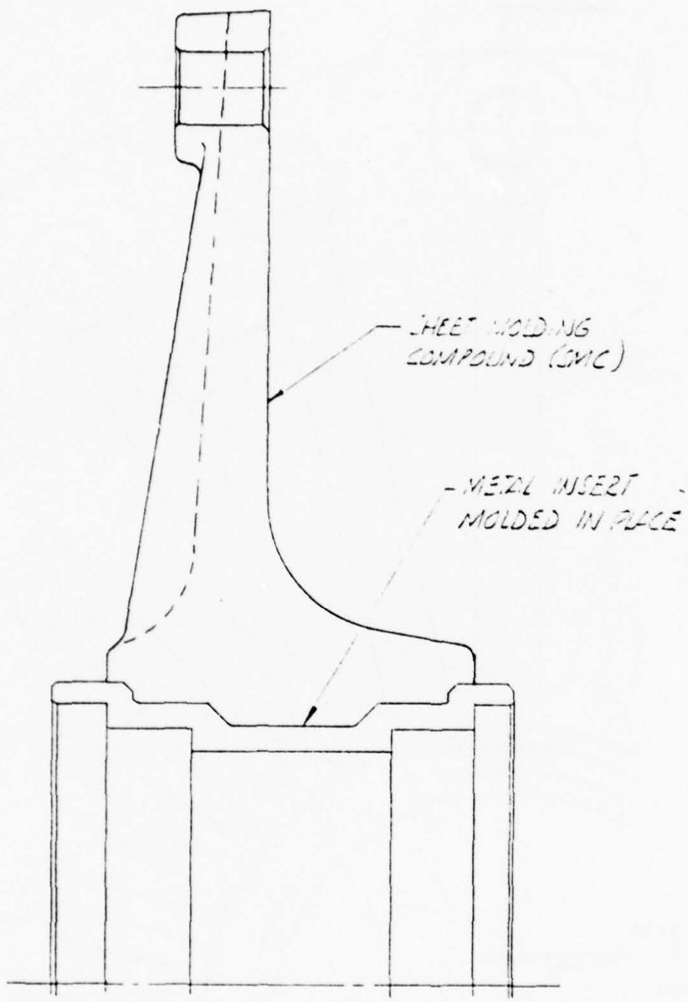


Figure 2-34 Cross Section of Hub

The stresses at the inner edge,

$$f^b = \frac{3 \cdot P_L}{2\pi \cdot t^2} \cdot K_1$$

and at the outer edge,

$$f^b = \frac{3 \cdot P_L}{2\pi \cdot t^2} \cdot K_2$$

where

$$K_1 = 1 - \frac{2 \cdot r_o}{r_o^2 - r_i^2} \left(\ln \frac{r_o}{r_i} \right)$$

$$K_2 = 1 - \frac{2 \cdot r_i^2}{r_o^2 - r_i^1} \left(\ln \frac{r_o}{r_i} \right)$$

with r_o and r_i designating the outer and the inner radius respectively and t the effective thickness of the plate.

The composite hub design has molded radial ribs at a spacing of 15°. The purpose of these ribs is to reduce the thickness of the basic plate. In order to be able to use the above formulas the ribbed cross section is replaced by a cross section of constant thickness with equivalent strength.

The moment of inertia was calculated for two sections and the minimum section moduli were determined. Then the constant thickness t' was calculated for an equivalent section modulus:

r in.	I in.^{4}	Z_{min} in.^{3}	t' in.
4.25	.1160	.1698	.957
7.50	.0579	.12886	.628

Then the radial stresses are:

$$K_1 = -.858$$

At r_i = 4.25 in., (see K_1 calculation for aluminum = -.858):

$$f^b = -16979 \text{ psi}$$

$$S.F. = \frac{35,000}{16,979} = 2.06$$

at r = 7.5 in, K_2 = .197

and the bending stress,

$$f^b = 14,112 \text{ psi}$$

and the safety factor,

$$S.F. \ \frac{35,000}{14,112} = 2.48$$

2. Wheel Mounting Holes - The maximum bearing load is 2,335 lbs. (limit) The maximum bearing pressure is:

$$f^{br} = \frac{3 \cdot 2,335}{2 \cdot .66 \cdot .88} = 6,030 \text{ psi}$$

3. Brake Drum Mounting Holes

Torque load per bolt,

$$P_B = \frac{83{,}000 \cdot 2}{16.50 \cdot 16} = 629 \text{ lbs/bolt}$$

Bolt bearing area $A_B = .477 \cdot .88 = .42 \text{ in.}^2$

Bearing pressure with all bolts loaded uniformly,

$$f^{br} = \frac{3 \cdot 629}{2 \cdot .42} = 2{,}245 \text{ psi}$$

The safety factor is large.

4. In-Plane Shear - The road loads are transmitted from the tire and rim to the axle by in-plane shear flow in the hub. The brake torque loads affect the hub only at the outer periphery.

For Condition 1 the shear flow is 355 lb/in (limit). The hub thickness at the mounting bolt radius is .45 in. The maximum shear stress is then,

$$f_1^s = \frac{335}{.45} = 744 \text{ psi}$$

Close to the hub at a radius of 4.25 in. the maximum shear flow for Condition 1 is $9{,}000/\pi \cdot 4.25 = 674$ lb/in., the basic hub thickness .70 in. and the maximum shear stress,

$$f_2^s = \frac{674}{.70} = 963 \text{ psi}$$

For Condition 2, where the brake torque is superimposed on the static load, the shear flow at the brake flange bolt radius is,

$$S = \frac{4{,}500}{\pi \cdot 8.25} + \frac{83{,}000}{8.25^2 \cdot 2\,\pi} = 367 \text{ lb/in}$$

The hub thickness is, .47 in. and the maximum shear stress,

$$f_3^s = \frac{367}{.47} = 782 \text{ psi}$$

Consequently the highest shear is 963 psi and the safety factor at the allowable in-plane shear stress of 7,500 psi.

$$S.F. = \frac{7{,}500}{963} = 7.8$$

5. Hub Center - The hub center transmits side loads, vertical loads and bending loads to the bearings. Because of the large areas in vertical and lateral directions, the compressive stresses between the molded hub and the metal bushing are obviously low.

However a check was made for bending stresses in the hub center flange caused by side loads.

For this purpose a conservative approximate analysis was used on a one inch wide radial slice of the hub at the bushing.

It was determined earlier that at a radius of 4.25 in. from the hub center the bending stress in 16,979 psi and the equivalent thickness is .957 in. The effective bending moment in this section is,

$$M_b = f^b \cdot Z$$

$$M_b = 2{,}592 \text{ in. lbs.}$$

This bending moment is resisted by a couple at the bearing centroids (2.2 in.) and results in a radial force of,

$$P_R = \frac{2{,}592}{2.2} = 1{,}178 \text{ lbs/in}$$

The maximum tension or compressive hoop stress resulting from the bending moment is approximately,

$$f = 1,178 = \frac{3.38^2 + 2.72^2}{3.38^2 - 2.72^2}$$

$$f = 5,508 \text{ psi}$$

For an allowable tension stress of 25,000 psi, the safety factor is large.

This analysis shows that all applied stresses in the wheel hub are acceptable and result in safety factors greater than 2.

2.7.2.5 Weight Analysis

The calculated weight of the composite wheel assembly excluding the hub is 33 lbs. The weight of the metal (steel) wheel is 124 lbs. The estimated weight savings is then 91 lbs. or 73%.

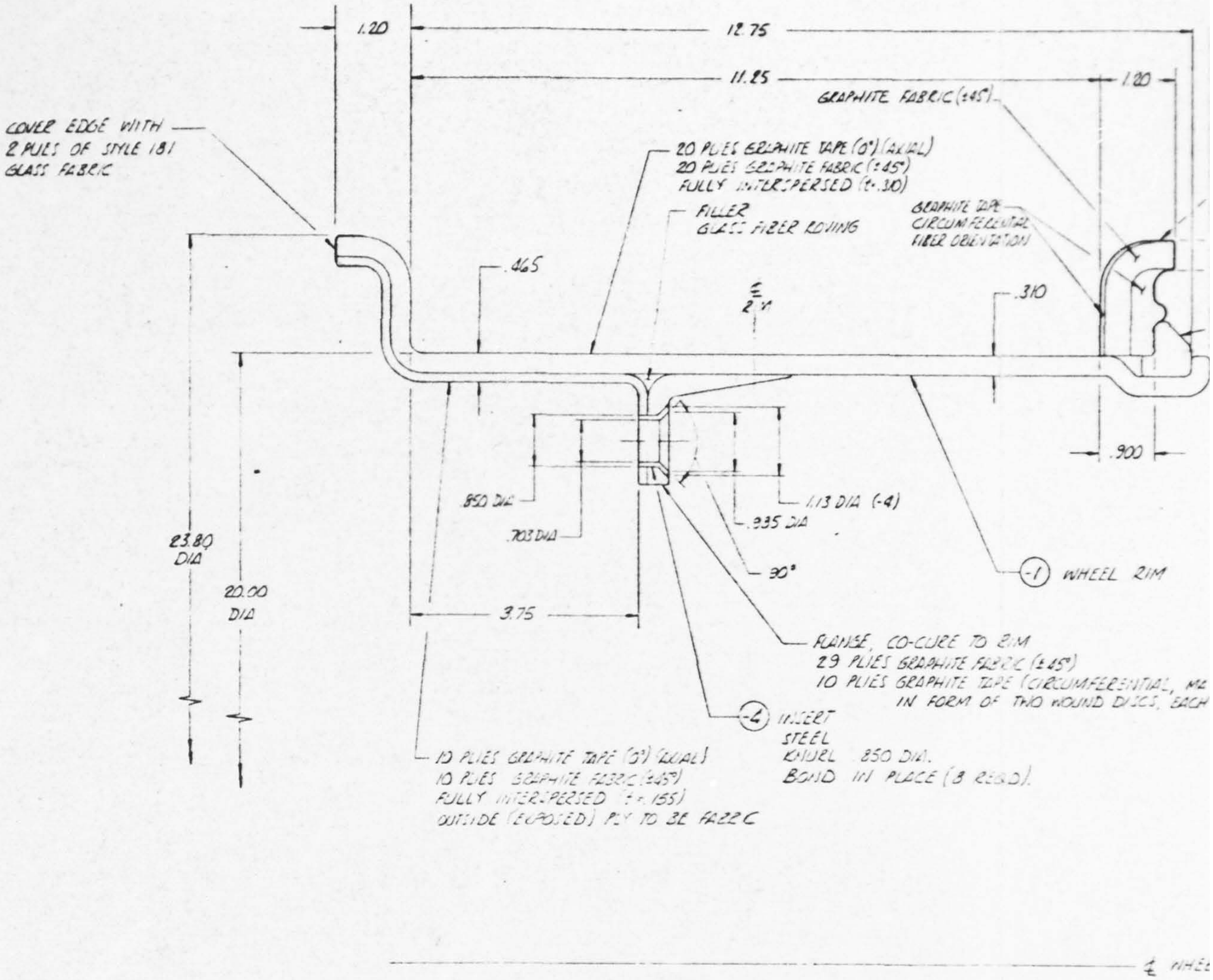
The calculated weight of the composite hub is 17.2 lbs. The weight of the metal (aluminum) hub is 20.9 lbs. The estimated weight savings is then 3.7 lbs. or 18%.

The weight savings per truck would be (for nine wheels) $(91 + 3.7) \cdot 9 =$ 852 lbs.

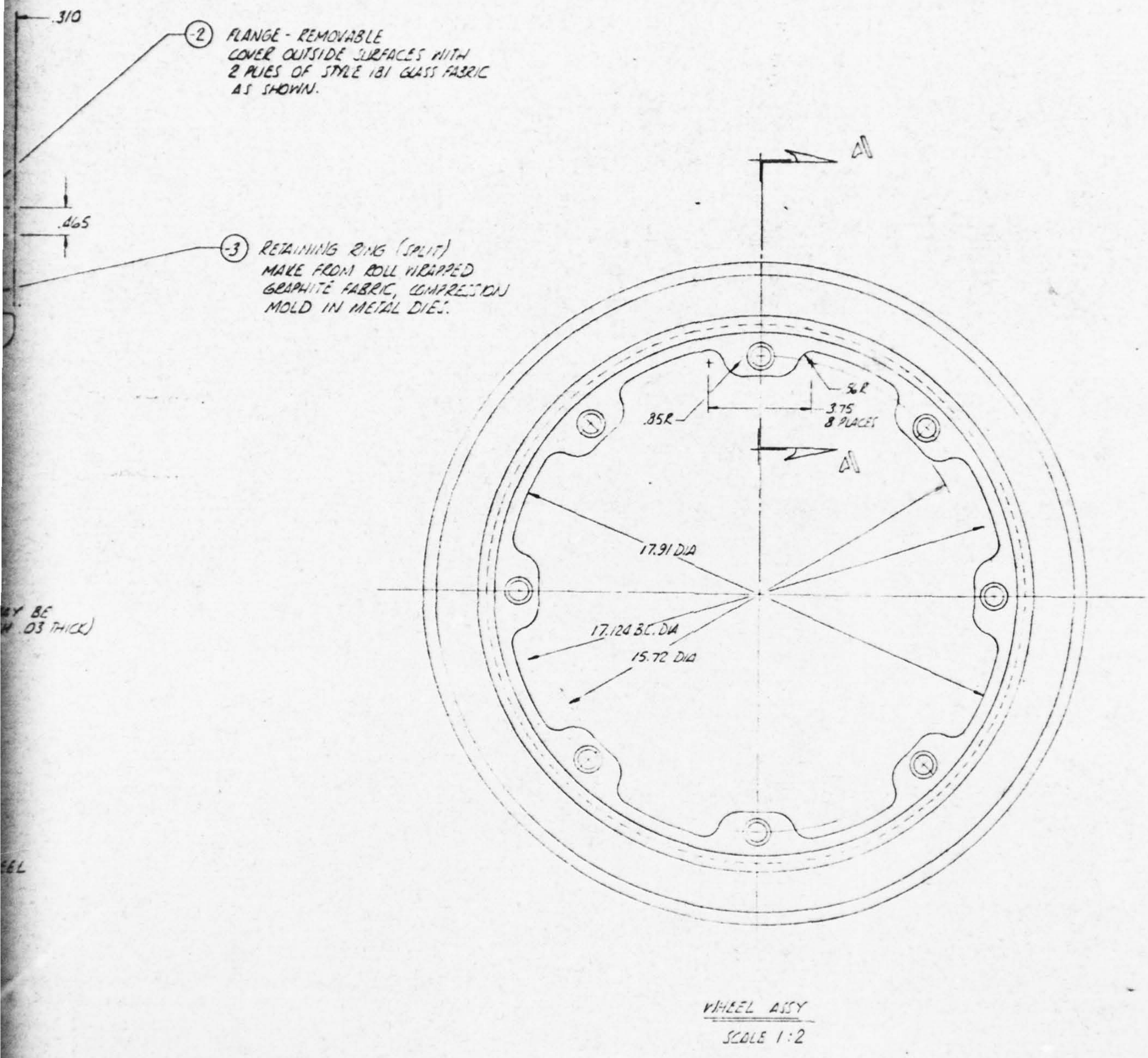
2.7.3 Manufacturing Analysis

A drawing of the truck wheel rim is shown on Figure 2-35 and of the hub on Figure 2-36. The rim is bolted to the hub to form a wheel assembly. The rim consists of a cylindrical rim with an integral inner flange for mounting to the hub and two outer flanges for tire retention. One of these flanges is integral with the cylinder, the other is removable in order to install the tire. The removable flange is held in place by a split ring which nests in a groove in the cylinder.

The layup and curing tool for the cylinder and flange is a segmented ring with the inside surface conforming to the outside contour of the wheel. The tool is segmented so that the rim can be removed from the tool after cure. The 40 plies which form the main body of the rim (See Figure 2-35) are laid up on the tool and precompacted. The 20 plies



SECTION A-A
SCALE 1:1



NOTES:

1. THIS DRA
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2. MATERIAL
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3. COMPOSITE
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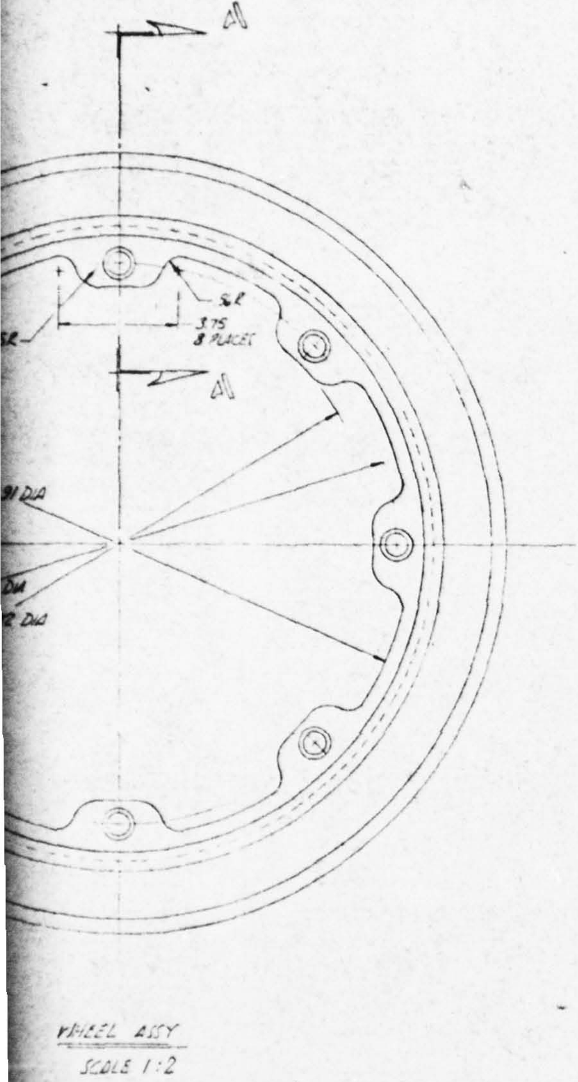
GLASS

Figure 2-35

Truck Wheel, C

QUANTITY REQUIRED	PART NO.	PART NAME
MANUFACTURING TOLERANCES ARE UNLESS OTHERWISE NOTED: DECIMALS .X ± 1" .XX ± .03 .XXX ± .010 ANGLES ± 1/2 DEG.	DRAFT E. Anderson 3-14-79	
ALL MACHINED SURFACES ARE AS NOTED REF. RMS TO	CHECK	
	DESIGN	
	STRESS	
	PROCESS	
	TOOLING	
	APPROVED	
NEXT ASSEMBLY / NO. REQ'D.		
CONTRACT NO.		

2



NOTES:

1. THIS DRAWING HAS BEEN PREPARED FOR COST ESTIMATING PURPOSES ONLY AND IS NOT FULLY DIMENSIONED. SCALE DRAWING FOR MISSING DIMENSIONS.
2. MATERIAL: GRAPHITE FIBER/EPOXY (TAPE AND FABRIC) AND GLASS FIBER FABRIC/EPOXY.
3. COMPOSITE MATERIAL NET WEIGHTS:
   GRAPHITE TAPE/EPOXY .... 10.8 LBS
   GRAPHITE FABRIC/EPOXY ...... 20.0 LBS
   GLASS FABRIC/EPOXY .... 2.5 LBS

5074

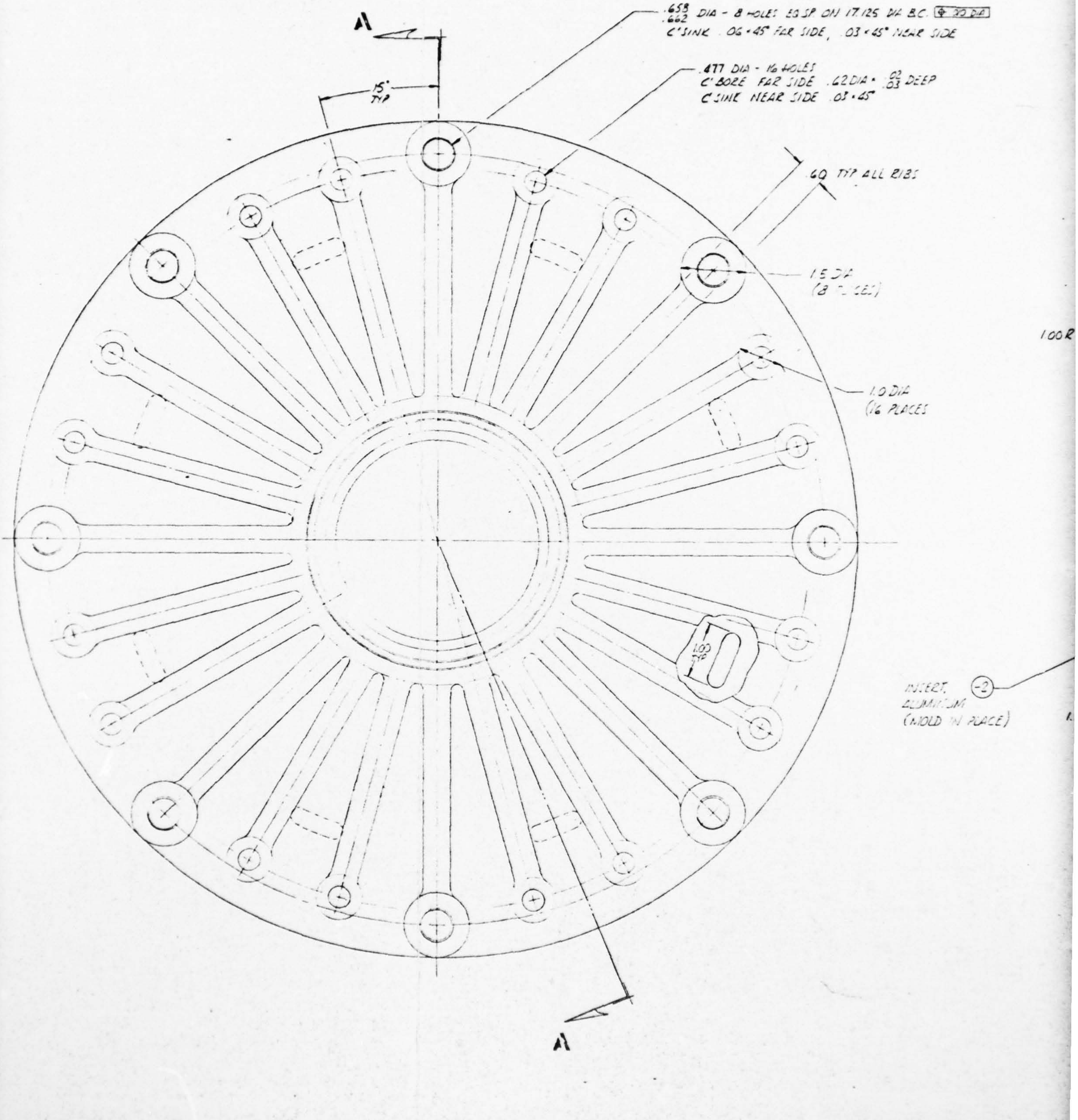
Figure 2-35

Truck Wheel, Composite Material

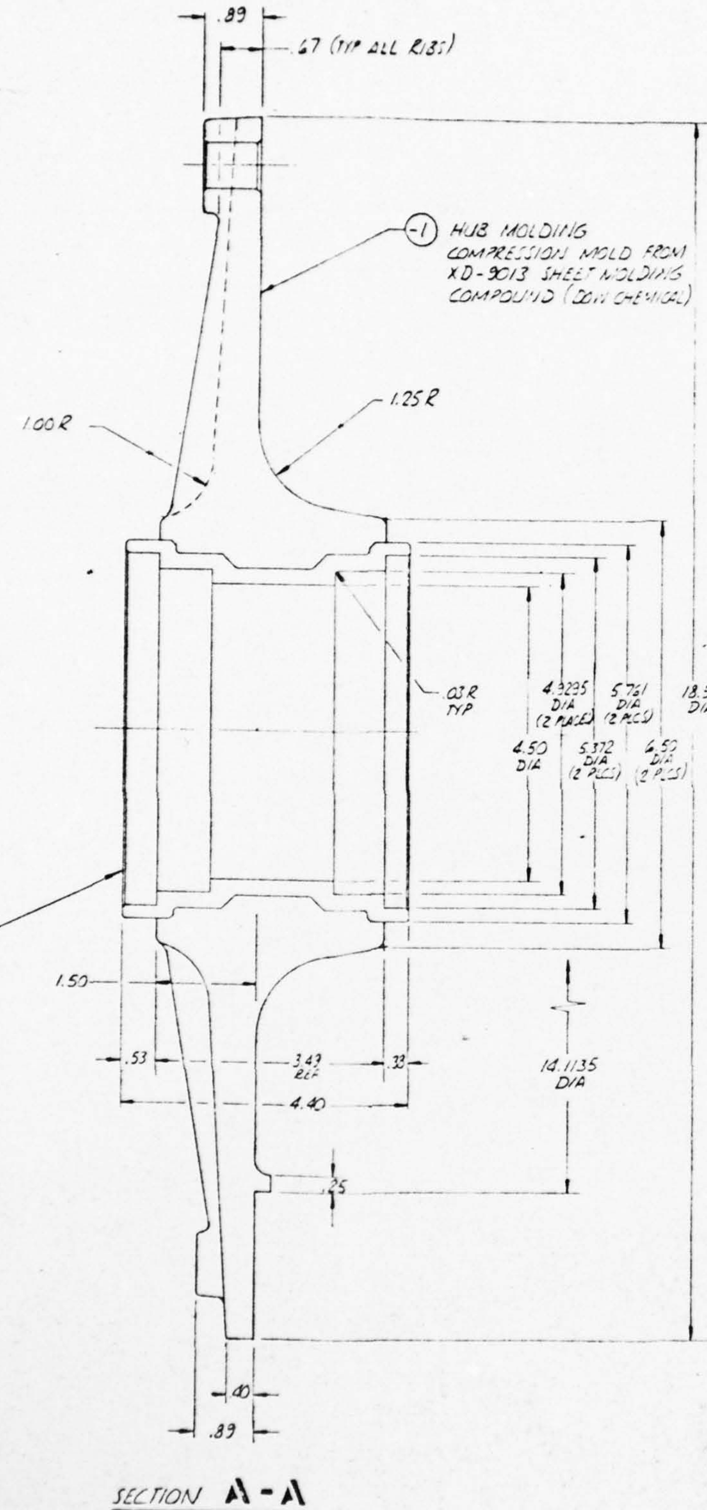
Riggs Engineering Corp.

WHEEL ASSY (TRUCK)
COMPOSITE MATERIAL (AMMRC)

DRAWING NO 5074



.658 .662 DIA - 8 HOLES EQ SP ON 17.125 DIA B.C.
C'SINK .06 × 45° FAR SIDE, .03 × 45° NEAR SIDE
.477 DIA - 16 HOLES
C'BORE FAR SIDE .62 DIA × .02 .03 DEEP
C'SINK NEAR SIDE .03 × 45°
A
15° TYP
.60 TYP ALL RIBS
1.5 DIA
1.0 DIA
(16 PLACES
1.00 TYP
INSERT,
(MOLD IN PLACE)
-2
1.00 R
A



NOTES:

1. THIS DRAWING HAS BEEN PREPARED FOR COST ESTIMATING PURPOSES ONLY AND IS NOT FULLY DIMENSIONED. SCALE DRAWING FOR MISSING DIMENSIONS.
2. COMPOSITE MATERIAL NET WEIGHT:
   XD-9013 SMC .... 15.3 LBS.
3. TOTAL WEIGHT: 17.2 LBS.

2

5075

Figure 2-36
Wheel Hub, Composite Material

QUANTITY REQUIRED	PART NO.	PART NAME	
MANUFACTURING TOLERANCES ARE UNLESS OTHERWISE NOTED:	DRAFT		Riggs Engineering Corp.
	CHECK		
	DESIGN		
	STRESS		HUB, WHEEL (TRUCK)
	PROCESS		COMPOSITE MATL (AMMRC)
	TOOLING		
	APPROVED		
NEXT ASSEMBLY / NO. REQD			SIZE R / CODE IDENT. NO. / DRAWING NO. 5075
CONTRACT NO.			SCALE 1/1 / CALC. WT / SHEET 1 OF 1

which form part of the cylinder, the inside and outside flanges are laid up on a rubber jacketed mandrel, precompacted and inserted in the rim tool. The 39 plies forming the rest of the internal flange is laid up and compacted on a separate tool. The outside contour after compaction must be slightly smaller than the inside diameter of the rim layup so that the flange can be pushed into the cylinder.

Curing pressure is provided through expansion of the rubber jacket on the mandrel for the left side of the wheel (Figure 2-35) and vacuum bag and autoclave pressure for the internal flange and the rest of the rim. The hard surface on the mandrel provides fixity for the internal flange location.

After curing it is necessary to machine the notched contour of the internal flange and the hole pattern. No other machining is required. The metal inserts in the holes are bonded in place.

The removable flange (Item 2 on Figure 2-35) is compression molded and cured in matched metal dies. The material is laid up as a flat ring, and stage formed prior to being placed in the die.

The retaining ring (Item 3 on Figure 2-36) is also compression molded and cured in matched metal dies. The material is formed by roll wrapping graphite fabric prepreg into a rod or tube, cutting it to length and placing it in the die. The split in the ring can be part of the die or the ring could be cut after curing at the bonded ends.

The hub shown on Figure 2-36 is compression molded from sheet molding compound. The aluminum insert in the center is molded in place. All features could be molded but it may be more desireable to machine the bearing seats and the locating tabs for the brake drums after molding to assure that the concentricity tolerances are held.

2.7.4 Manufacturing Costs

The design of the truck wheel rim shown on Figure 2-35 is suitable for production in quantities less than 500. For larger production quantities as pointed out in the discussion of the drive wheel in paragraph 2.3.4, further studies in fabrication techniques involving continuous fiber composites must be conducted and new low cost methods must be developed in order to produce a cost effective part.

The wheel hub shown in Figure 2-36 is molded from sheet molding compound. This process is suitable for large as well as small production quantities.

Estimated costs to manufacture the wheel rim and the hub are shown in Table 2-14 for quantities 10, 100 and 10,000 units.

Table 2-14 Truck Wheel Manufacturing Costs

Truck Wheel Rim (Figure 2-35)

Development Engineering and Manufacturing Drawings $4500.

Quantity	10	100	10,000*
	Cost/Unit, $		
Material	1801	1525	1091
Fabrication and Quality Control	1578	1313	778
Prod. Engineering and Program Adm.	978	140	9
Tooling	449	55	4
Total:	4806	3033	1882

Wheel Hub (Figure 2-36)

Development Engineering and Manufacturing Drawings $1000.

Quantity	10	100	10,000*
	Cost/Unit, $		
Material	90	72	69
Fabrication and Quality Control	323	191	109
Prod. Engineering and Program Adm.	129	40	5
Tooling	1288	209	8
Total:	1830	512	191

*Fabrication Rate: 1000 units/year

3.0 Cost and Weight Comparison

A comparison of the estimated costs of the composite material components with the actual costs of the present metal components is shown in Table 3-1. The costs for the composite components are based on a quantity of 10,000 which was reported in the previous paragraphs under Manufacturing Costs. The costs for the metal components are taken from the Specifications and Requirements section of the contract. No cost figure was given for the truck wheel.

The comparison shows that for most components the composite parts are considerably more expensive to manufacture than their metal counterparts. This is due in part to the fact that in a development program the composite part is required to be interchangeable with the metal part and therefore the envelope and shape are restricted, making it difficult to utilize the fibrous material to its full potential. Another reason is the lack of low cost manufacturing methods for large quantity production. To date application of advanced fibrous composite materials to structural components of complex shapes have been mainly in the aerospace industry where production quantities are comparatively small. More studies are required in the area of fabrication techniques in order to produce cost effective composite materials parts.

Table 3-2 shows a comparison of weights between the composite components and the present metal parts. The largest weight saving is for the drive wheel which is also the most difficult and most expensive to fabricate. Considerable weight saving is also achieved for the truck wheel, especially the rim assembly. Fabrication of the rim assembly is relatively difficult but simpler than the drive wheel hub. The truck wheel hub is a compression molding which is quite simple to manufacture. The existing metal hub is made of aluminum so the weight saving is less than for the steel components. Costs were not given for the metal truck wheel.

The small weight saving shown for the Idler Wheel and Road Wheel is in part due to the fact that the rubber wheel must be replaceable and requires special arrangement. Also the metal hub is made of aluminum. More weight could be saved on the road wheel if the requirement of interchangeability with the idler wheel were deleted. The design is dictated by the load conditions which are much more severe on the idler wheel than on the road wheel.

Table 3-1

Cost Comparison

Component	Cost, $ Metal	Composite
1. Torsion Bar	98.85	
2. Drive Wheel		
Hub	597.77	2598
Sprocket	179.33	1016
3. Track Support Roller	27.75	239
4. Track Idler Wheel	150.87	1068
5. Track Road Wheel	150.87	1068
6. Track End Connector Link	3.50	-
7. Truck Wheel		
Rim Assy.	-	1878
Hub	-	195

Table 3-2

Weight Comparison

Component	Weight, Lbs. Metal	Composite	Weight Savings lbs.	%
1. Torsion Bar	105	-	-	-
*2. Drive Wheel (Incl. Sprockets)	523	158	365	70
3. Track Support Roller	22	15.7	6.3	29
4. Track Idler Wheel	105	102.7	2.3	2.2
5. Track Road Wheel	105	102.7	2.3	2.2
6. Track End Connector Link	2.6	-	-	-
**7. Truck Wheel (Incl. Hub)	144.5	50.2	94.3	65
* Drive Wheel (Hub)	303	113.3	189.7	63
(2 Sprockets)	220	44.6	175.4	80
** Truck Wheel (Rim Assy.)	123.6	33	90.6	73
(Hub)	20.9	17.2	3.7	18

4.0 Prototype Development

The projected costs to carry out development to one prototype part for each of the components is shown below. The costs are based on the design concepts shown in the report and it is assumed that little or no additional engineering is required other than producing prototype manufacturing drawings.

Drive Wheel

Hub (Figure 2-17)

Design Engr. and Drawings	$ 1,000.
Material	4,870.
Fabrication and Q.C.	3,100.
Process Engr. and Proj. Adm.	1,470.
Tooling	16,100.
Total:	$26,540

Sprocket (Figure 2-18)

Design Engr. and Drawings	$ 500.
Material	1,316.
Fabrication and Q.C.	780.
Process Engr. and Proj. Adm.	1,040.
Tooling	8,970.
Total:	$12,606.

Support Roller (Figure 2-19)

Design Engr. and Drawings	$ 500.
Material	110.
Fabrication and Q.C.	490.
Process Engr. and Proj. Adm.	250.
Tooling	4,490.
Total:	$5,840.

Idler and Road Wheel (Figure 2-26)

Design Engr. and Drawings	$ 1,000.
Material	1,230.
Fabrication and Q.C.	1,000.
Process Engr. and Proj. Adm.	500.
Tooling	12,880.
Total:	$16,610.

Truck Wheel

Wheel Rim (Figure 2-35)

Design Engr. and Drawings	$ 1,000.
Material	1,800.
Fabrication and Q.C.	2,370.
Process Engr. and Proj. Adm.	500.
Tooling	4,490.
Total:	$10,160.

Hub (Figure 2-36)

Design Engr. and Drawings	$ 500.
Material	90.
Fabrication and Q.C.	485.
Process Engr. and Proj. Adm.	250.
Tooling	12,880.
Total:	$14,205.